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REPORTS, PAPERS, DISCUSSIONS, AND MEMOIRS

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FOURTH PROGRESS REPORT OF THE SPECIAL COMMITTEE TO REPORT ON STRESSES IN RAILROAD TRACK*

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December 23, 1924

* Presented to the Annual Meeting, January 21, 1925.

CONTENTS

I.—Introduction

To the American Society of Civil Engineers:

The Special Committee to Report on Stresses in Railroad Track herewith presents its fourth progress report.

I.—Introduction

1.—Preliminary.—As has been the case since its organization in 1914 the Committee has co-operated with the Special Committee on Track of the American Railway Engineering Association, the membership of the committees of the two Societies being almost identical, and this report is presented simultaneously to the two Societies, and also to the American Railway Association, which has furnished financial support and has otherwise co-operated in the work of the investigation.

The first progress report of the Committee was published in Vol. LXXXII of the *Transactions* of the American Society of Civil Engineers (1918); it may also be found in Vol. 19 of the *Proceedings* of the American Railway Engineering Association. The second progress report was published in Vol. LXXXIII of the *Transactions* of the American Society of Civil Engineers (1919-20), in Vol. 21 of the *Proceedings* of the American Railway Engineering Association, and as *Circular No. S-II-10* of the American Railway Association. The third progress report was published in Vol. LXXXVI of the *Transactions* of the American Society of Civil Engineers (1923), Vol. 24 of the *Proceedings* of the American Railway Engineering Association, and as *Circular No. IV-32* of the American Railway Association. In the three progress reports, comprising in all about 520 pages, the tests that have been made on railroad track and in the laboratory to give information on the action of rails, ties, and ballast under the loads of locomotives are described and the results compared and discussed. The data and discussion of a great variety of tests are to be found in the reports given in these *Transactions* and *Proceedings* of the Societies.

The work herein reported includes tests on straight track and curved track of the electrified portion of the Chicago, Milwaukee and St. Paul Railway in Montana, with several types of electric locomotives, one steam locomotive, and loaded freight cars, reported as "II.—Tests on the Chicago, Milwaukee and St. Paul Railway", and tests on straight track and curved track of four Eastern railways with steam locomotives and loaded cars under the heading, "III.—Tests on Eastern Railroads", which were made principally to get information on the influence of canting of rail by the use of inclined tie-plates. With the great amount of material obtained in the field work it was not possible, in view of the limitations of space, to report all the data of the tests, and an effort has been made to choose for publication those parts which have the most direct bearing on the problems considered and which also may be useful for purposes that may develop hereafter.

The Committee is continuing work on the subject assigned to it.

2.—Acknowledgment.—Since the third progress report was issued, the expenses of carrying on the work of the Committee have been paid largely

from contributions made by the American Railway Association. The Committee expresses its appreciation of this support. Important contributions have also been received from companies that manufacture steel rails, these funds being handled through the American Society of Civil Engineers and the American Railway Engineering Association. The Committee expresses appreciation of the contributions made by the Illinois Steel Company, the Carnegie Steel Company, the Tennessee Coal, Iron and Railway Company, the Algoma Steel Corporation, the British Empire Steel Corporation, the Cambria Steel Company, the Bethlehem Steel Company, the Colorado Fuel and Iron Company, the Midland Steel Company, and the Westinghouse Electric and Manufacturing Company.

The co-operation of railroads in furnishing facilities for the test work has itself been an important contribution. The Chicago, Milwaukee and St. Paul Railway, C. F. Loweth, Past-President, Am. Soc. C. E., Chief Engineer, and L. K. Sillcox, M. Am. Soc. C. E., General Superintendent of Motive Power, gave excellent opportunities for the conduct of the test work by providing locomotives, track, train crews, and other facilities throughout the six weeks occupied by the tests. L. J. Murray, Special Apprentice, gave valuable assistance throughout the tests, and others of the Engineering, Mechanical, and Operating Departments were very helpful. On the Eastern work, the Baltimore and Ohio Railroad, Earl Stimson, M. Am. Soc. C. E., Chief Engineer, Maintenance, the Reading Company, R. B. Abbott, Assistant General Superintendent, the Lehigh Valley Railroad, G. L. Moore, Engineer, Maintenance of Way, and the Richmond, Fredericksburg and Potomac Railroad, E. M. Hastings, M. Am. Soc. C. E., Chief Engineer, all furnished adequate facilities for making the tests conducted on the track of these several railroads. In the tests on both the Chicago, Milwaukee and St. Paul Railway and the Eastern railroads much interest was taken and valuable assistance rendered by those in charge of operation.

The field and office work has been carried on by a regular staff well fitted for this class of work. E. E. Cress, Assistant Engineer of Tests, in charge of field and office work and the reduction of data and preparation of material, is entitled to much credit; his thorough familiarity with details, keen grasp of the problem, and helpfulness in the study and preparation of the data have made his services particularly valuable. Louis J. Larson, Assoc. M. Am. Soc. C. E., Associate in Theoretical and Applied Mechanics in the University of Illinois, by reason of his special training and insight in testing work, has given valuable service in both laboratory and field work during the summer seasons. Mr. R. Ferguson has given helpful service in the field work, the reduction of data, and the preparation of the report, as did Mr. H. L. Parr in the field work. Others have assisted in the work from time to time and all have given loyal and careful service.

The University of Illinois has continued to co-operate in the work by giving the use of laboratory, shop, and office facilities, and through the service of the members of the staff of the Engineering Experiment Station from time to time.

II.—Tests on the Chicago, Milwaukee and St. Paul Railway

3.—The Tests.—The tests on the Chicago, Milwaukee and St. Paul Railway were conducted in July and August, 1923, on the electrified section of the railroad near Lennep and Loweth, Mont., stations which are on the Rocky Mountain Division 35 and 45 miles west of Harlowton. A purpose of the tests was to obtain data by which the effects on straight track and on medium and sharp curves produced by the several types of electric locomotives run at several speeds could be judged and compared with each other and with a standard type of steam locomotive. Stresses in the rail under and between wheels and for both vertical and lateral bending of the rail were determined and various other observations bearing on the action of the rail and the locomotives were made.

4.—The Track.—The track on which the tests were conducted was the single-track main line. The test section of straight track was within the yard limits of Lennep about 2 500 ft. east of the station building and alongside a passing track. The gradient was 0.79% downward to the east. The test section of 6° curve was at Loweth near the telegraph office and a short distance eastward from the summit of the pass over the Belt Mountains. The gradient was 0.79% downward to the east. A passing track lay alongside. The test section of 10° curve was about 2 300 ft. west of the Lennep station building. Its gradient was 0.75% downward to the east. The 6° curved to the left and the 10° to the right when going eastwardly, the direction in which the test runs were made. All three test sections were on a slight fill, which was solid and well compacted.

The rail was 90-lb. A. R. A.-A. That on the straight track was laid in 1918, on the 6° curve in 1916, and on the 10° curve in 1921. The rail on the straight track and 6° curve was only slightly worn. On the 10° curve the outer rail was worn $\frac{1}{8}$ in. or more on the gauge side; there was a small lip on the inner rail. The moment of inertia of the full section about a horizontal gravity axis is 38.7 in.^{4} and that about the vertical gravity axis, 7.5 in.^{4}. The corresponding section moduli, $\frac{I}{c}$, are 15.2 in.^{3} about the horizontal axis and 2.9 in.^{3} about the vertical—both being with respect to a fiber at the base of rail. As the wear of the rail affects the value of the section modulus with respect to the base of rail only slightly, the values of $\frac{I}{c}$ for the full rail were used in the calculations to determine bending moments from stresses in rail. The splice-bars were 24-in., heat-treated angle-bars of A. R. A.-A. design.

The tie-plates were $6\frac{1}{2}$ by 9 in. on the straight track and 10° curve, and $6\frac{1}{2}$ by $8\frac{1}{2}$ in. on the 6° curve. The eccentricity of the former (distance from center of length of tie-plate to center of base of rail) was $\frac{3}{16}$ in. and of the latter $\frac{5}{16}$ in. The tie-plates were approximately flat.

The ties were 7 by 8 in. by 8 ft. untreated fir and except for renewals had been in the track eight years. The ties at the test locations were in fairly good condition except a few on the 6° curve. On the straight track the spacing was 18 to the rail length of 33 ft., and 20 on the two curves.

The ballast was mainly gravel. On straight track it extended 17 in. below the bottom of the tie and on the 10° curve, 15 in. On the 6° curve there were 12 in. of mixed gravel, cinders, and sand. In all cases the lower part of the ballast was good but the upper part was in poorer condition.

The maintenance work for the season of 1923 had been deferred until after the tests were made so as not to change conditions, and the track was thus not up to the usual standards of the railroad. Generally it was in only fair surface and alignment. At the test location on straight track the alignment was fairly good, though on each side there were stretches with poor alignment. The surface was somewhat uneven. In spots tamping was needed. A number of the tie-plates next to the joints were found to be somewhat loose. The 6° curve, a point where the speed of trains was generally slow, was in poorer condition, the inner rail being uneven and a number of the tie-plates under that rail being loose. At the test section the alignment was good. The 10° curve was in very good condition, holding close to a 10° curvature, except that the alignment over a pile bridge 200 ft. west of the test section was poor and that of the easement at the beginning of the curve was not good; other than at these points the locomotives rode well on this curve to a point beyond the test section. The condition of the track in the three locations is described at this length in order that it may be judged whether conditions of the track over the approach to the stretches at which the instruments were located would be likely to affect the action of the locomotives and cars in their passage over the test sections, particularly at the higher speeds.

The gauge on the test section and adjoining it was 4 ft. 8½ in. on straight track, 4 ft. 8¾ in. on the 6° curve, ranging from 4 ft. 8⅝ in. to 4 ft. 9⅛ in. on the 6° curve, and 4 ft. 9 in. on the 10° curve with little variation. The super-elevation of the outer rail of the 6° curve was 3.2 in., corresponding to a speed of 29 miles per hour, and there was little variation from this value. On the 10° curve the super-elevation was 3.9 in., corresponding to a speed of 24 miles per hour. The maximum speeds used in the tests, 50 miles per hour on the 6° curve and 40 miles per hour on the 10° curve, were more than one and two-thirds times the speeds corresponding to the super-elevation.

The data from measurements of the depression of the rail made under the weight of a loaded car and of a light caboose and used in the calculation of the value of the modulus of elasticity of rail support, u, by the method described in previous progress reports, gave an average value of 1 100 lb. per in. for the straight track.

5.—The Locomotives and Cars.—Three principal types of electric locomotive were in regular use on the electrified portions of the railroad: (*a*) the General Electric freight locomotive; (*b*) the General Electric passenger locomotive; and (*c*) the Westinghouse-Baldwin passenger locomotive. The last type was represented in the tests by three forms, the original form of the locomotive and two modifications. The five electric locomotives used in the tests then represented all the types of motive power used for main-line service on the electrified divisions of the railroad. The steam locomotive was represented by one of the Mikado type. The General Electric freight locomotive

has been described in detail.* A description of the General Electric and Westinghouse-Baldwin passenger locomotives of the Chicago, Milwaukee and St. Paul Railway may be found in two articles published by the American Institute of Electrical Engineers.† These articles and information received from the railroad company form the principal basis for the descriptions which follow.

The three principal types of electric locomotive differ widely in both the mechanical and the electrical features of their design. A common characteristic was that the axles of the drivers were individually driven, separate motors being applied to the driving axles, and there were no connecting rods or side rods for which counterbalancing must be provided, and, therefore, no stresses in the track due to imperfect counterbalance. All the electric locomotives were symmetrical with respect to the center of their length.

The power supply was direct current of 3 000 volts.

In Figs. 1 and 2 are diagrams of the locomotives—wheel loads and spacings and other data. A diagrammatic representation of the equalization system is given in Figs. 3 and 4. Additional data will be found in Table 1.

The General Electric freight locomotive, 10221, had been in freight service since 1916; it does not differ from the other freight locomotives in use on the electrified divisions. The characteristics of this type, as affecting the action on the track, are the unusually long wheel spacings, the great over-all length, and the fact that a spring gear drive with axle suspension motors is used. The suspension of the motors puts on each driving axle an unsprung weight of 16 250 lb., a greater unsprung load per axle than is found in either of the newer types. Each half of the locomotive has a four-wheel guiding truck similar to the ordinary type used on steam locomotives, and two trucks, each containing two pairs of drivers. There is a hinged connection between the two halves, which also permits some vertical movement, but no horizontal movement other than a swing. The two sections of the cab are supported on bolsters set midway between pairs of driving axles; the center plate on the bolsters allows longitudinal and pivoting movement, but no lateral displacement of the cab.

The General Electric passenger locomotive, 10254, is of the type used between Othello and Tacoma, Wash., and had been in service since 1920. The locomotive is driven by bi-polar gearless motors having the armatures wound on the axles. Other characteristics of the locomotive, as affecting the track, are the large number of drivers, which are very closely spaced, and the relatively small diameter of the drivers and the relatively small loads they carry. The close wheel spacing has an important effect in the development of smaller stresses in the rail than would occur with a longer spacing. The first two of the twelve drivers on one side are grouped in a truck frame with a guiding wheel, then follows a unit of four drivers; the second half is the reverse of this. The running gear is thus articulated in three places. The cab is divided into three sections and supported in an unusual manner. The end sections contain auxiliary equipment, switches, resistances, and the com-

* *General Electric Review,* November, 1916, p. 929.
† *Journal,* Am. Inst. Elec. Engrs., April, 1920.

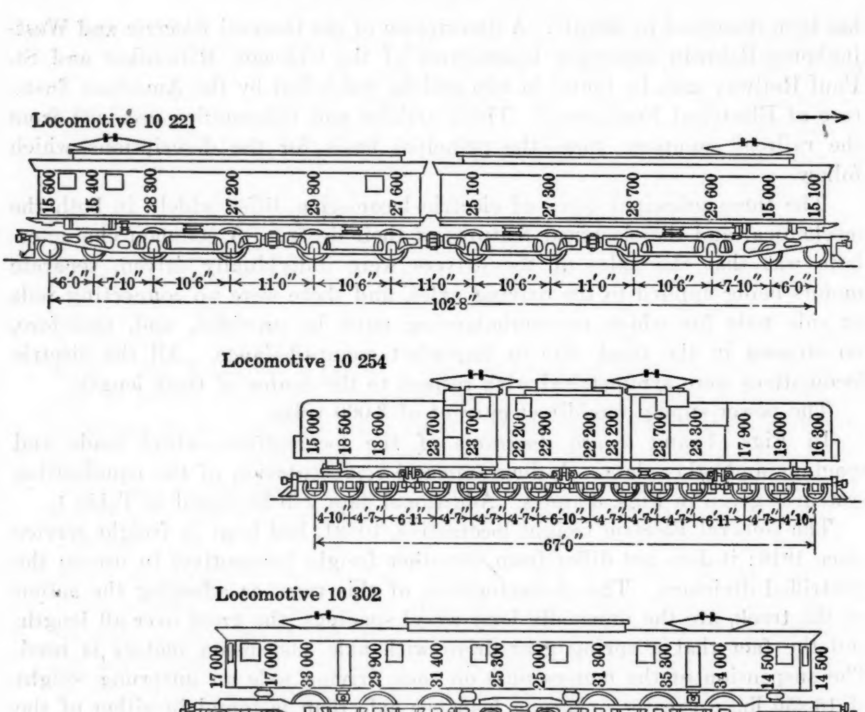


FIG. 1.—DIAGRAMS OF ELECTRIC LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

partments for the crew. These end sections are fastened rigidly at the center of each middle set of drivers and rest on rollers bearing on inclined planes over the end sets of the running gear. The center section containing the train heating and lighting equipment is suspended from four brackets on the adjacent ends of the other two sections and has no other connection with the remainder of the locomotive to give it support. The guiding axle at either end of the locomotive is allowed some lateral play but the movement is damped by wedges placed above the journal boxes. Other details bearing on the action of the cab and running gear in traversing curves can not well be given. Having the motor armatures on the axles places a considerable proportion of the total axle load below the springs; the results of the tests thus have some bearing on the problem of the effect of the unsprung weight on the track. The unsprung or dead weight per driving axle is given as about 9 600 lb. In order to have one of this type for use in the tests, this locomotive was brought from the Coast Division, a distance of 600 miles, including the intermediate non-electrified divisions of the railroad.

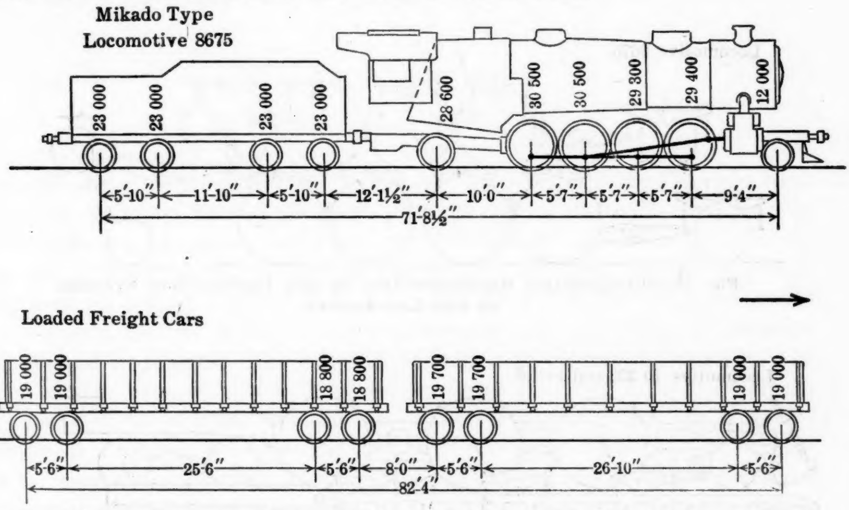


FIG. 2.—DIAGRAMS OF STEAM LOCOMOTIVES AND CARS OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

The Westinghouse-Baldwin locomotive, 10302, is of the type being operated between Harlowton, Mont., and Avery, Idaho, and had been in service since 1920. This locomotive is essentially two Pacific running gears coupled back to back, supporting a single cab that extends the whole length of the locomotive. It has "Woodward" four-wheel guiding trucks and "Rushton" trailers. The coupling between halves of the running gear consists of a long bar with a pin at each end. The cab is supported by six spring plungers on each half of the running gear, two being in line with the center pin and two at each end. The driving torque is transmitted from twin armature motors (this weight being spring supported) to a quill drive and thence to the individual driving axles. Since they were put into service, the weight of all the Westinghouse-Baldwin locomotives has been increased by the substitu-

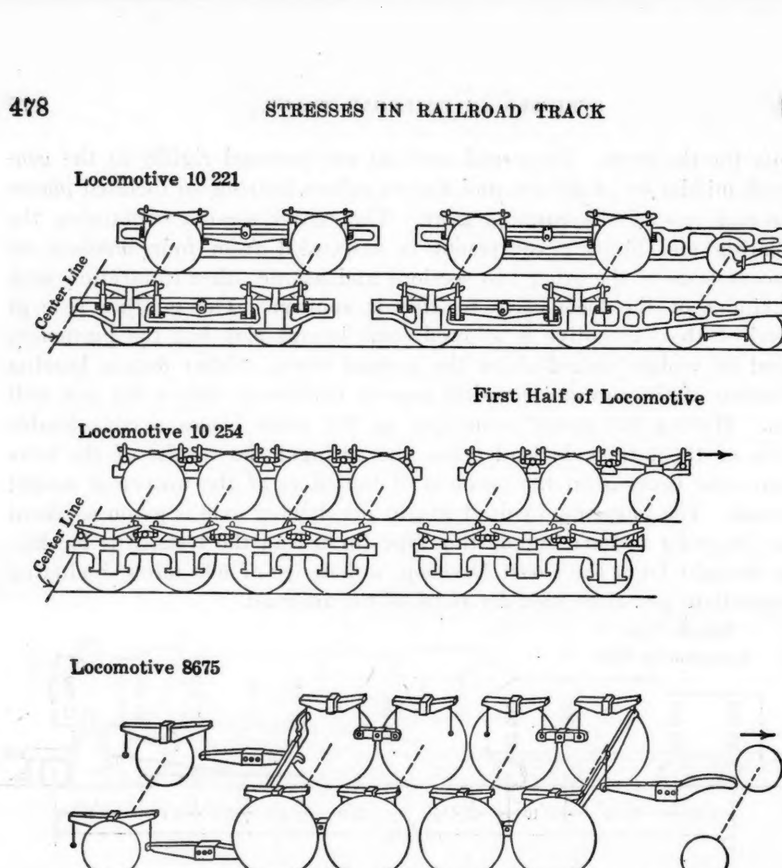


FIG. 3.—DIAGRAMMATIC REPRESENTATION OF THE EQUALIZATION SYSTEMS OF THE LOCOMOTIVES.

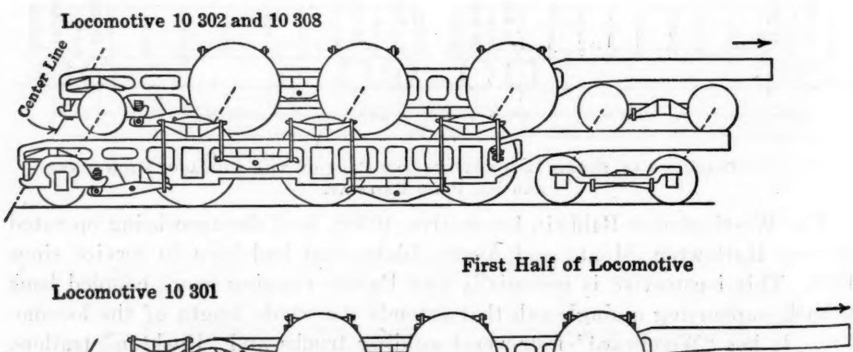


FIG. 4.—DIAGRAMMATIC REPRESENTATION OF THE EQUALIZATION SYSTEMS OF THE LOCOMOTIVES.

tion of heavier frames, by additional bracing for the cabs, and by changes in the quill drives, so that the wheel loads reported in the diagrams and tables are somewhat higher than those named in the earlier descriptions of the locomotives.

TABLE 1.—Data of the Three Types of Electric Locomotives of the Chicago, Milwaukee and St. Paul Railway.

	General Electric Freight Locomotive 10 221.	General Electric Passenger Locomotive 10 254.	Westinghouse-Baldwin Passenger Locomotive 10 302.
Total weight of locomotive, in pounds	569 225	547 700	617 500
Total weight on drivers, in pounds	447 100	485 300	390 700
Average load per driving wheel, in pounds.	28 000	20 200	32 600
Unsprung weight per driving wheel, in pounds	8 100	4 800	3 900
Diameter of driving wheels, in inches	52	44	68
Length over all, in feet	112.0	76.0	88.6
Height of center of gravity, in inches	62.6	60	71.5
Number of motors	8	12	6*
Total horse-power, 1-hour rating	3 440	3 500	4 200
Total horse-power. continuous	3 000	3 200	3 400
Total tractive effort, continuous, in pounds.	70 700	42 000	49 000
Corresponding speed, in miles per hour	15.9	28.4	26.0

*Twin armature motors.

Modifications had been made in two of the Westinghouse-Baldwin locomotives in an effort to find whether improved lateral flexibility and gain in maintenance would result, and these two locomotives were also used in the tests. In Locomotive 10308 the cab was given a different support, the ends being held on a long vertical link or pendulum; a different connection was made between the two halves of the running gear, but the "Rushton" trailers were allowed to remain. The cab of Locomotive 10301 was divided into two parts, and the over-all length increased about 10 ft. Two four-wheel trucks were placed in the middle, replacing the trailers or forming two sets of double trailers. A link was placed in this truck of the second half of the locomotive to improve its guiding action. The addition of four wheels decreased the weight on the drivers somewhat.

Locomotive 8675 is a Mikado type freight locomotive known as the heavy U. S. Railroad Administration design (Type L 3 of the Chicago, Milwaukee and St. Paul Railway) and had been in service since 1919. It had been run 20 000 miles since shopping. It was brought in from the Musselshell Division at Miles City, Mont., for use in the test. The weight on a trailer is nearly as great as that on a driver. The total load on the drivers* was 239 000 lb. The same design is in use on railroads in various parts of the United States.

In comparing the locomotives, it is seen that the driving wheel base was 10 ft. 6 in. on the General Electric freight locomotive, 13 ft. 9 in. on the General Electric passenger locomotive, and 16 ft. 9 in. on the Westinghouse-Baldwin passenger locomotives and the Mikado type. Even the last two are

* The unsprung weight, including the wheel and axle and bearings, a proportion of the connecting rod and side-rods, but not the effect of counterbalance at speed, is estimated to be 6 000, 6 500, 10 000, and 6 000 lb., respectively, for a first, second, third, and fourth driver.

considerably shorter than the wheel base of 22 ft. found with Santa Fé type locomotives. The diameter of the drivers of the General Electric freight locomotive was 52 in., that of the General Electric passenger locomotive, 44 in., that of the Westinghouse-Baldwin locomotive, 68 in., and that of the Mikado type, 63 in. The locomotives were taken direct from service without change or special preparation. All of them may be said to have been in ordinarily good condition mechanically.

Although the electric locomotives were designed to run in either direction, for the tests generally the test runs were made eastward with the following named end in front: 10221, B; 10254, A; 10302, No. 2; 10308, No. 1; and 10301, No. 2. It was the practice to turn some of these locomotives at the end of each run for a certain mileage, then to reverse their direction for a time.

The freight cars were Chicago, Milwaukee and St. Paul Railway coal cars that had seen much service. They were loaded with coal so as to give nearly equal weights on the two ends. The weight was taken for each truck of the loaded car and then the whole car was weighed as a check. The wheel loads given in Fig. 2 are one-fourth of the load on one end of the car. The wheels were 33 in. in diameter, and the wheel base, 5 ft. 6 in.

6.—Conduct of Tests and Reduction of Data.—The methods used in the tests of track were generally the same as those used in the tests described in the previous progress reports. Eight stremmatographs were used simultaneously. These instruments had been rebuilt with some minor improvements over the form described in the first progress report.* Four stremmatographs were placed on one rail between ties at distances apart of about 6 ft., and the other four instruments directly opposite on the other rail. The driving mechanism used rotated the disks of all the instruments simultaneously. The correlation of a point on the record of one of the disks with the point of any other disk at the same moment was possible. As the locomotive passed the test section a record of the strains in the rail was made on each instrument. The passage of each wheel of one side of the locomotive thus was recorded on four instruments, and each wheel of the other side on four other instruments. As in each instrument one disk recorded the strain at one edge of the base of rail and another that at the other edge, the differences in the stresses at the two edges, and, therefore, the lateral bending stresses, were obtainable. Three or four runs were recorded on a disk, according to the length of the locomotive. The disks were then taken out of the instruments and fresh ones inserted.

Except in the motoring-regenerating tests made with two locomotives in action, the power was shut off as the locomotive approached the test section of track and the locomotive "coasted" past the instruments. The speeds were read from a speedometer in the cab connected with the tread of a wheel, the instrument having been checked up over a measured length of track. The locomotives were easily controlled to give the speed desired. After a run, the locomotive was backed over the track and the next run made. The speeds

* *Transactions*, Am. Soc. C. E., Vol. LXXXII (1918), p. 1224.

used in the tests ranged from 5 to 60 miles per hour on straight track, 5 to 50 miles per hour on the 6° curve, and 5 to 40 miles per hour on the 10° curve. The sequence was usually 5, 25, 40, and 60, or 5, 30, 40, and 50, or 5, 25, and 40 miles per hour. About sixteen runs were made at each test location for each speed with the General Electric freight locomotive, the General Electric passenger locomotive, and the original form of the Westinghouse-Baldwin locomotive, and about eight runs with the other forms of Westinghouse-Baldwin locomotive and the Mikado type locomotive and cars.

Fig. 5 shows the position of the stremmatographs in one of the tests of curved track. Variations from this arrangement were made, but the diagram is representative of all the tests. As the track was laid with alternate joints it was necessary that a rail joint lie within the test section.

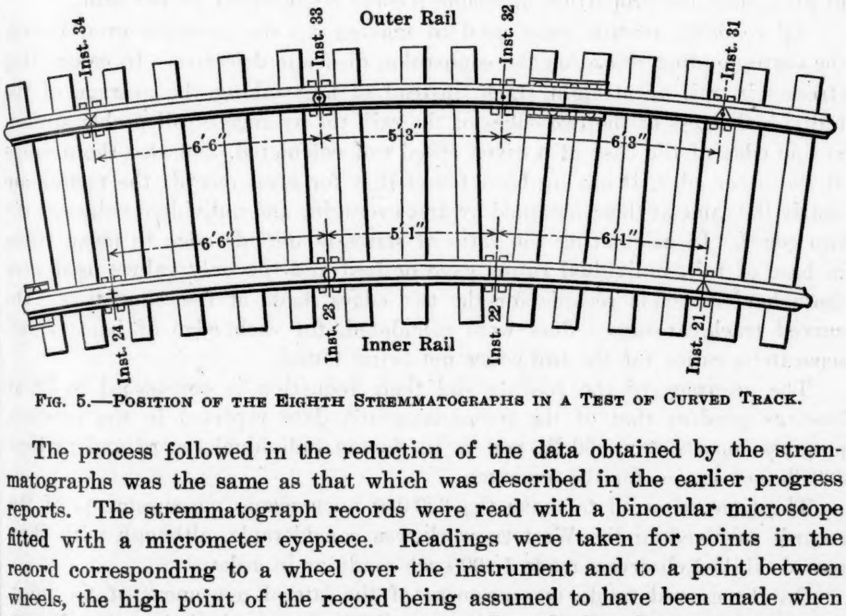


FIG. 5.—POSITION OF THE EIGHTH STREMMATOGRAPHS IN A TEST OF CURVED TRACK.

The process followed in the reduction of the data obtained by the stremmatographs was the same as that which was described in the earlier progress reports. The stremmatograph records were read with a binocular microscope fitted with a micrometer eyepiece. Readings were taken for points in the record corresponding to a wheel over the instrument and to a point between wheels, the high point of the record being assumed to have been made when the wheel was directly over the instrument. In reducing these measurements the readings were multiplied by the proper microscopic constant and then reduced to stresses by multiplying by a constant which involves the position of the neutral axis of the rail section, the vertical distance of the needle-bar below the base of rail, the modulus of elasticity of steel (taken as 30 000 000 lb. per sq. in.), and the gauge length (which was 4 in.). A correction was also made to allow for the variation of moment and stress over the gauge length to obtain the maximum stress at the middle of the gauge length by multiplying the average stress over the gauge length for wheel over instrument by the factor, 1.04, as was done with the tests recorded in the first progress report, so that the stresses reported at points of positive moment are stresses in pounds per square inch at the base of the rail at the middle of the gauge length. The variation over the gauge length at points of

negative moment was slight and no correction was used for the stresses at such points.

The records on the disks were found to be unusually good, that is, readable and definite. As must be expected some of these were indefinite and not usable. On the straight track, those obtained at all but the speed of 60 miles per hour generally were clear, probably 85% were usable. At the speed of 60 miles per hour, the proportion available was much less, the jar from the rail affecting the clearness of the record; particularly for the two instruments on one rail adjacent to the joint less than half the records were usable. On the 6° curve the proportion of usable records at 50 miles per hour was considerably less than at the lower speeds, and the outer rail gave better records than the inner rail. The records on the 10° curve were generally excellent at all speeds, the proportion of usable records being about 95 per cent.

All readable records were used in making up the averages even though the corresponding record on the companion disk was defective. In calculating stresses in rail on straight track, instead of first taking the average of the individual stress at the two edges of the rail, the average of all usable stresses at one edge of the base at a given speed was calculated, and also the average at the other edge, it having been found that for good records the results are nearly the same as those obtained by first averaging the individual values at the two edges. In calculating the ratio of stress at outside edge to mean stress in base of rail, individual ratios were desired and the only values used were those having usable records for the two edges made at the same time. On curved track, average values were calculated, for each edge of base of rail separately, ratios for the two edges not being found.

The accuracy of the records and their reduction is considered to be at least as good as that of the stremmatograph data reported in the previous progress reports, say, 800 lb. per sq. in. for an individual record and perhaps 300 lb. per sq. in. for the averages.

The time required to make the 320 000 microscopic measurements of the records involved in the Western work was considerable, although with clear records skilled observers made 1 200 such readings in a day.

On the curved track, measurement of the lateral movement of the inner and the outer rail, as the locomotive moved by at a speed of about 2 miles per hour, was made. A wooden bar with an Ames dial gauge at one end had one end attached to a fixed point 5 ft. from the rail and the other against a point on the head of the rail, the dial being read as the wheel passed and at a point midway between wheels. Measurement was made of the position of the flanges of the wheels of the locomotive after it came to rest on the two curves without the application of brakes, and also of the alignment of the axles and the amount of the journal play under the same condition. The contour of the tires and of the rails was also obtained.

During the latter part of the time given to the field work a method was developed for determining the position of the bearing of the wheel on the rail as it passed by at any speed. A copper wire placed under one rail and over the other was kept taut and pulled a distance of about 6 in. between the

passage of one wheel and the next, one man pulling the wire and another at the other side of the track keeping steady tension in the wire. Near the end of the field tests, it was found that a series of punch marks in a line along the head of the rail would give impressions on the wire that enabled the position of the flattened wire to be determined accurately with respect to the width of the head of the rail and thus the bearing of wheel on rail to be known. If the flange of the wheel passed quite close to the rail an impression by it was also made. The wire used was that available; the diameter ranged from 0.06 to 0.10 in.

7.—*Results of Tests on Straight Track.*—The tests on straight track give information on the stresses in rail developed by the six types of locomotives with their great variety of wheel spacings and loadings, of positions of load, and of arrangement of springs and equalizers; they also provide a basis for a comparison with the results on curved track.

In Tables 2 and 3 are given the values of the mean stresses in base of rail under each wheel at three or four speeds for the five electric locomotives, each value being the average of the records of the eight instruments on the two rails. The mean stresses in base of rail developed by the Mikado type locomotive are given in Table 4. In Figs. 6 to 12 the values of the mean stress in base of rail are plotted to scale and also the mean stress for points between wheels. It should be noted that each value is the average of a large number of observations and that individual observations will be either greater or less than the average value.

It will be seen that the stresses developed in the 90-lb. rail under the drivers at a speed of 5 miles per hour average about as follows: General Electric freight locomotive (10221), 17 000 lb. per sq. in.; General Electric passenger locomotive (10254), 8 000 lb. per sq. in; Westinghouse-Baldwin passenger locomotives (10302, 10308, and 10301), 15 500 lb. per sq. in., ranging from 13 500 to 19 700 lb. per sq. in. under the several drivers; and Mikado type locomotive (8675), 12 000 lb. per sq. in., varying from 10 000 to 14 000 lb. per sq. in., under the several drivers. With the exception of the General Electric passenger locomotive, which developed smaller stresses in the rail, these stresses are within the general range of stress noted in the previous progress reports for the heavy steam locomotives with 90-lb. rail on straight track at the low speeds. In the discussion of the effect of speed on stress in rail that is given in Article 8 which follows, it is shown that a change of speed from 5 to 40 and 60 miles per hour results in an increase of stress in rail that is generally less than was found in the tests previously reported.

In the first progress report a method* was developed for calculating the stresses in the rail produced by any arrangement of static wheel loads on a track with a given rail and known stiffness of track. The effect of individual wheels, both for the rail under the wheel and for points away from the wheel, was determined from a formula which involved the wheel load, the moment of inertia of the rail, and the stiffness of the rail supports (ties and ballast) and from a master diagram that shows the change in bending

* *Transactions*, Am. Soc. C. E., Vol. LXXXII (1918), p. 1202; *Proceedings*, Am. Ry. Eng. Assoc., Vol. 19, p. 882.

TABLE 2.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK WITH GENERAL ELECTRIC LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

(Stresses are given in pounds per square inch.)

LOCOMOTIVE 10 221

Speed, in miles per hour.	Rear End Truck. 2.	Rear End Truck. 1.	Driver. 8.	Driver. 7.	Driver. 6.	Driver. 5.	Driver. 4.	Driver. 3.	Driver. 2.	Driver. 1.	Head End Truck. 2.	Head End Truck. 1.
5	9 900	6 900	17 100	16 800	17 100	17 400	16 800	16 900	17 000	16 200	7 400	9 800
25	10 300	7 200	17 200	17 400	17 200	17 600	16 900	17 300	17 100	16 700	7 700	10 400
40	10 600	7 600	18 000	17 900	18 500	18 300	17 400	18 000	17 700	16 800	7 900	10 800

LOCOMOTIVE 10 254

Speed, in miles per hour.	Rear End Truck. 2.	Rear End Truck. 1.	Driver. 12.	Driver. 11.	Driver. 10.	Driver. 9.	Driver. 8.	Driver. 7.	Driver. 6.	Driver. 5.	Driver. 4.	Driver. 3.	Driver. 2.	Driver. 1.	Head End Truck. 2.	Head End Truck. 1.
5	8 100	.....	7 400	8 000	8 700	7 600	7 800	8 900	7 900	7 400	7 400	8 700	8 000	7 700	.....	8 900
25	7 900	.....	7 200	7 500	8 400	7 300	7 600	8 700	7 600	7 400	7 300	8 500	7 700	7 400	.....	9 000
40	8 500	.....	7 700	6 100	8 500	7 700	8 000	8 500	8 200	7 800	7 400	8 700	8 400	7 700	.....	8 700
60	9 200	.....	9 200	9 200	10 000	9 300	9 200	9 500	10 000	10 000	9 100	9 400	9 400	8 800	.....	9 300

moment away from the wheel. As an illustration of the effect of wheel spacing and load, calculations of stresses have been made by this method for Locomotives 10221 and 10254, and the values of the calculated stresses in the rail have been plotted as shaded lines in Figs. 6 and 7 besides the observed stresses at 5 miles per hour. A comparison of the two sets of values shows that the stresses and their distribution are quite similar, probably as nearly the same as are the nominal and the real weights. For Locomotive 10254, the calculated stress at points between wheels, for negative moments, is somewhat greater than the observed stress when the space between wheels is large; a smaller but opposite effect is found for the closer spacing. The two diagrams (Figs. 6 and 7) call attention to the smaller stresses that are found with smaller individual wheel loads and a favorable spacing of the wheels, a wheel load of 28 000 lb. in Locomotive 10221, producing a stress of 16 000 lb. per sq. in. and one of 22 000 lb. in Locomotive 10254 producing only 8 000 lb. per sq. in. The stress under the drivers of Locomotive 10254 is only about 50% of that due to an isolated wheel load, while that under the drivers of Locomotive 10221 is 85 per cent. As noted in the first progress report, the effect of wheel spacing will depend somewhat upon the weight of rail and stiffness of track. It is evident that wheel spacing and distribution of load are among the factors that need consideration in locomotive design.

TABLE 3.—Mean Stress in Base of Rail on Straight Track with Westinghouse-Baldwin Passenger Locomotives of the Chicago, Milwaukee and St. Paul Railway.

(Stresses are given in pounds per square inch.)

Speed, in miles per hour.	Rear End Truck.		Driver.			Trailer.		Driver.			Head End Truck.	
	2.	1.	6.	5.	4.	2.	1.	3.	2.	1.	2.	1.
					Locomotive 10 302							
5	10 500	6 800	13 800	15 300	15 200	10 700	7 800	13 900	19 700	13 500	5 300	9 300
25	10 700	7 000	14 100	15 500	15 300	10 700	8 200	14 000	20 500	13 800	5 500	9 900
40	11 500	7 400	14 800	16 100	15 800	11 200	8 800	14 500	20 800	14 800	6 600	10 700
60	13 900	10 100	17 000	18 100	17 600	13 500	11 200	16 400	22 100	15 700	8 600	11 400
					Locomotive 10 308							
5	9 200	5 800	14 000	14 700	14 900	9 600	8 200	15 100	16 300	15 300	6 200	10 000
40	9 800	6 700	14 800	16 000	15 400	10 200	8 500	15 300	16 500	14 200	6 600	10 700
60	10 200	7 700	15 600	17 500	16 200	10 800	9 400	16 800	17 600	14 800	8 000	11 300
					Locomotive 10 301							
5	8 200	5 200	14 100	15 900	15 600	7 000 5 900	5 800 6 400	16 300	14 400	14 100	5 800	9 800
40	8 300	5 500	14 200	15 700	15 800	6 900 5 700	6 600 6 200	15 500	14 100	14 000	6 100	9 500
60	8 700	6 500	16 100	17 100	16 600	7 600 6 700	7 100 6 900	16 900	16 000	14 800	6 900	10 200

The method of calculating stresses referred to has been found to apply well to all track used in the various tests made by the Committee. The reversed process has been found to be a useful way of ascertaining the wheel loads, or, at least, of checking reported weights, especially as scales for weighing individual loads are not common. As was described in the third progress

report,* an algebraic equation is written out for the vertical bending stress for each driver (or, instead, the equivalent single wheel load); this equation involves the wheel load for a given wheel and those of the adjacent wheels, all represented as unknown quantities, and the stress in the rail under the given wheel as a known quantity. There will then be as many equations and unknown quantities as there are wheels. The solution of these equations is tedious, but not troublesome. The values found for the individual wheels may be changed proportionately to make the sum of the loads agree with the total load of the locomotive, but the correction so made has generally been within 4 per cent. In Fig. 13 are given the nominal weights and the loads calculated from the stresses in rail by this method. The agreement between the weights reported and the calculated wheel loads for 5 miles per hour is generally fairly good. In three of the locomotives a difference of more than 6 000 lb. in a wheel load was noted, with several variations at other wheels ranging from 2 000 to 4 000 lb. Generally, the calculated load on the truck wheels and trailers was greater than the reported load.

TABLE 4.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK WITH MIKADO TYPE LOCOMOTIVE OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

(Stresses are given in pounds per square inch at base of rail.)

Speed, in miles per hour.	Position of counterweight.	Trailer.	Driver. 4.	Driver. Main.	Driver. 2.	Driver. 1.	Front truck.
5	Mean value	15 000	14 100	12 400	9 700	12 600	8 400
25	Mean value	15 500	14 900	13 300	10 000	13 200	9 100
Increase in stress due to speed in percentage of stress at 5 miles per hour...............		3	6	7	3	5	8
40	Up	......	10 500	15 500	8 500	10 000	.....
	Down	......	20 500	11 500	12 500	17 500	.....
	Mean value	16 200	15 600	13 400	10 500	13 700	9 700
Increase in stress due to speed in percentage of stress at 5 miles per hour................		8	11	8	8	9	16
Increase in stress due to counterbalance in percentage of stress at 5 miles per hour ...		......	35	16	19	30	.....

In connection with a discussion given subsequently it should be noted that at the place where the tests on straight track were made there was a siding at one side of the main track and with this and the accompanying conditions the support possibly was somewhat different for the two rails. However, on the whole, there was found to be little difference in the action of the two rails; the summation of the stresses in the two rails for all the wheels of the six locomotives at a speed of 5 miles per hour differed only 2 or 3 per cent. Generally, too, the stresses under wheels of the same axles did not differ greatly, indicating that the axle load was evenly divided between the wheels. In a few cases the stress under one wheel was 10 to 15% greater than the mean of the stresses for the companion wheel. The two loaded cars gave stresses uniformly higher on the left rail by about 8% of the mean of

* *Transactions*, Am. Soc. C. E., Vol. LXXXVI (1923), p. 1013; *Proceedings*, Am. Ry. Eng. Assoc. Vol. 24, p. 397.

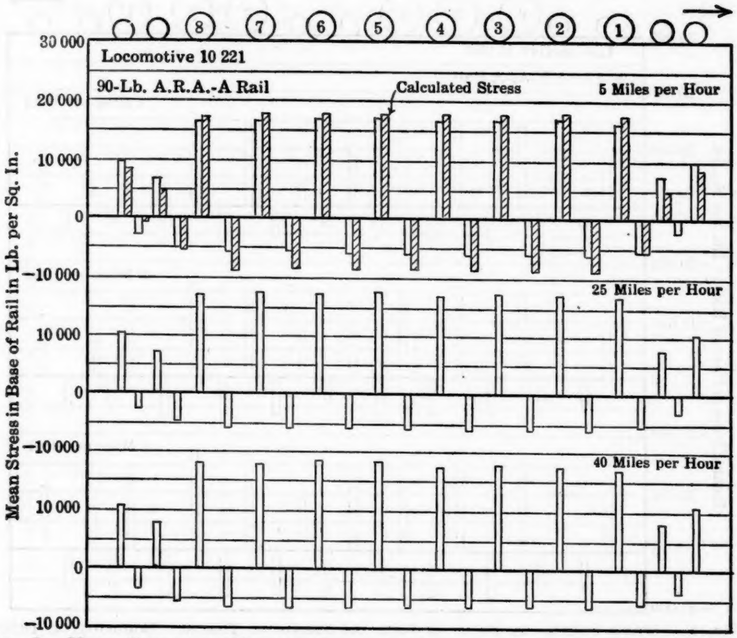


FIG. 6.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOCOMOTIVE 10221 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

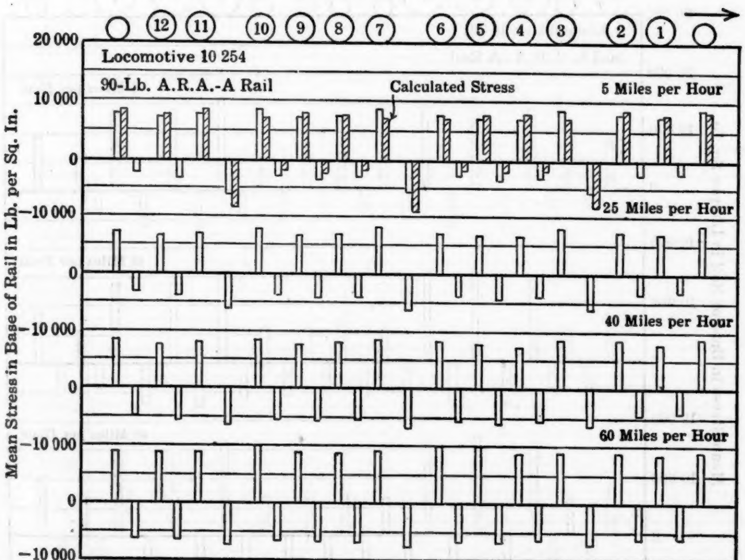


FIG. 7.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOCOMOTIVE 10254 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

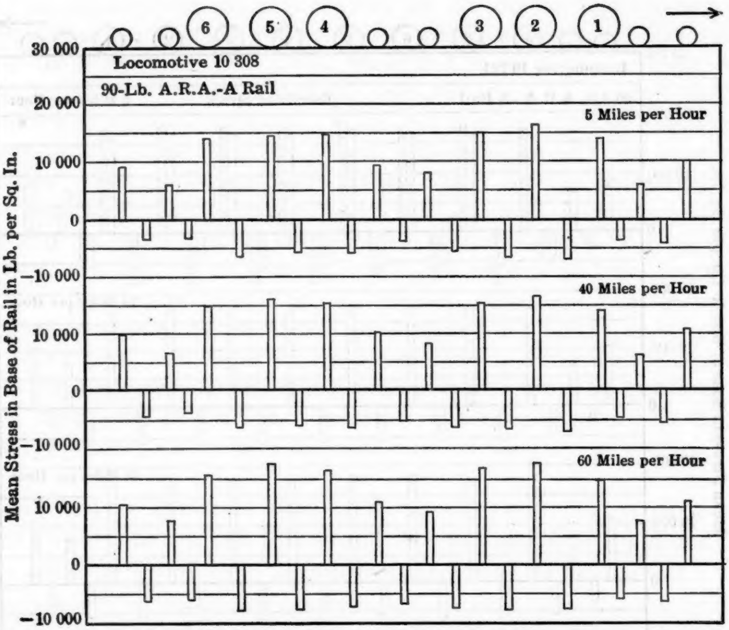


FIG. 8.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOCOMOTIVE 10308 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

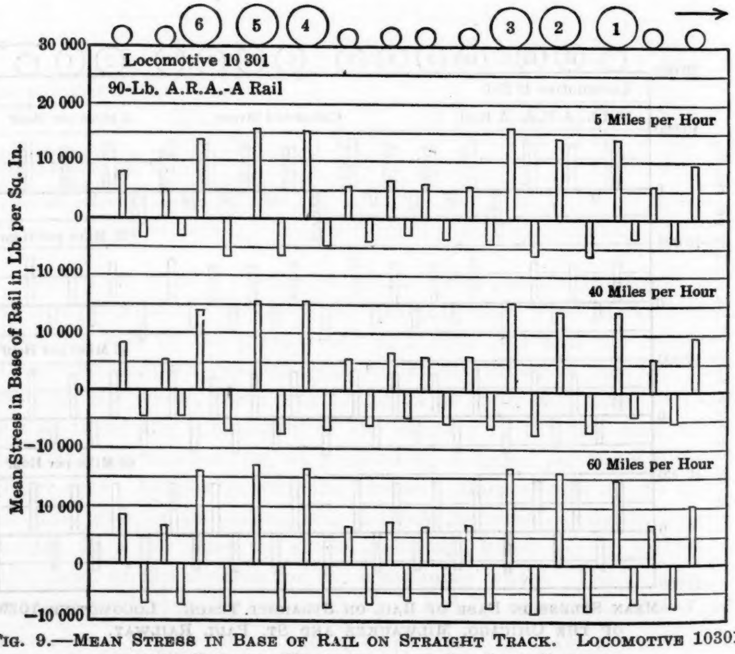


FIG. 9.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOCOMOTIVE 10301 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

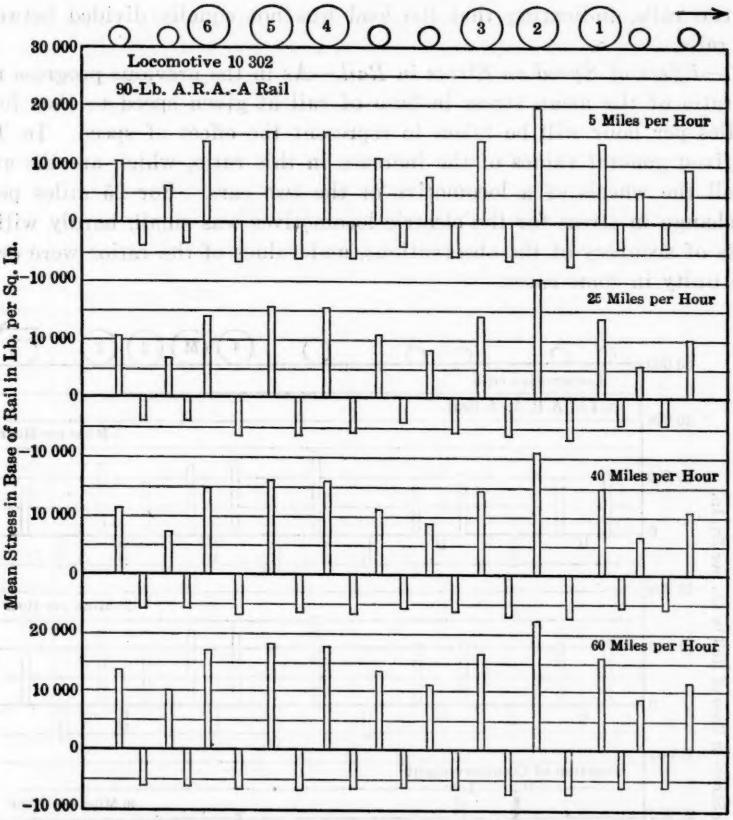


FIG. 10.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOCOMOTIVE 10302 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

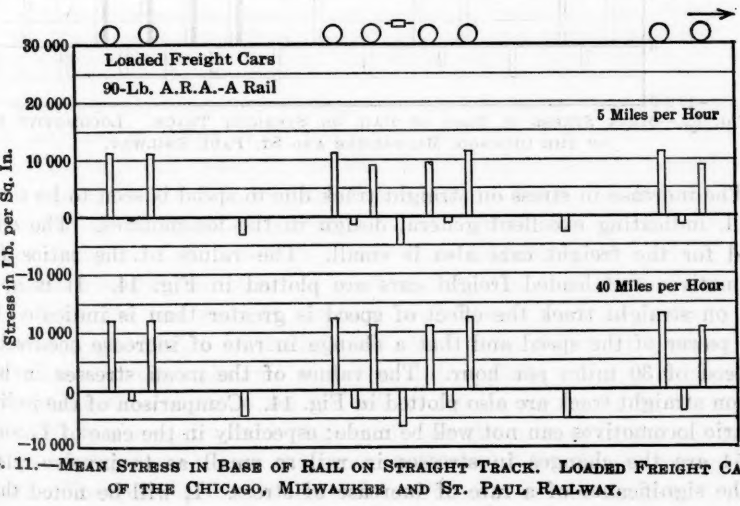


FIG. 11.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOADED FREIGHT CARS OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

the two rails, indicating that the load was not equally divided between the two rails.

8.—Effect of Speed on Stress in Rail.—As in the previous progress reports, the ratio of the mean stress in base of rail at given speed to that found at 5 miles per hour will be taken to represent the effect of speed. In Table 5 are given general values of the increase in this ratio, which are the averages for all the wheels of a locomotive or the two cars. For 25 miles per hour the change in stress for the electric locomotives was small, hardly within the limits of accuracy of the observations, and values of the ratios were even less than unity in some cases.

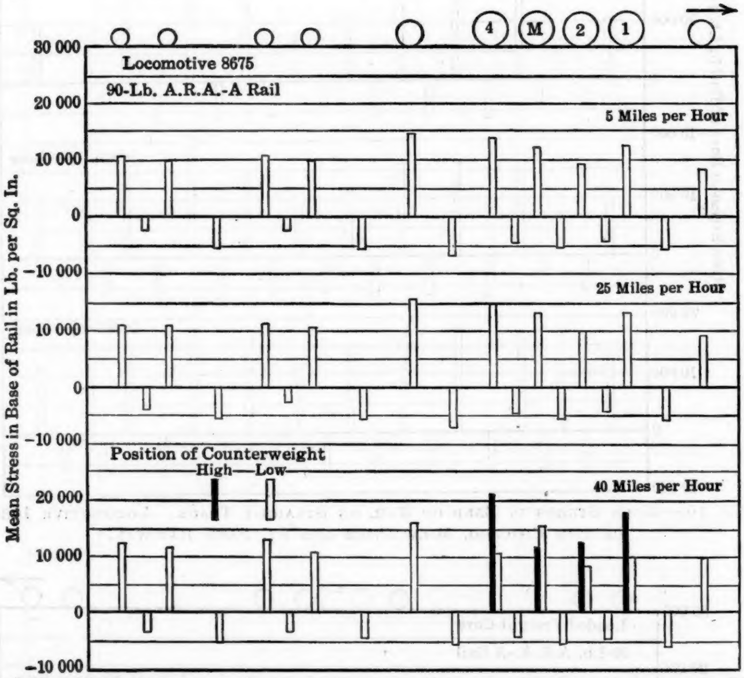


FIG. 12.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOCOMOTIVE 8675 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

The increase in stress on straight track due to speed is seen to be relatively small, indicating excellent general design in the locomotives. The effect of speed for the freight cars also is small. The values of the ratios for the locomotives and loaded freight cars are plotted in Fig. 14. It is apparent that on straight track the effect of speed is greater than is indicated by the first power of the speed and that a change in rate of increase occurs beyond a speed of 30 miles per hour. The values of the mean stresses in base of rail on straight track are also plotted in Fig. 14. Comparison of the individual electric locomotives can not well be made; especially in the case of Locomotive 10254 are the changes in stresses in rail so small as to involve questions of the significance of a rate of increase of stress. It will be noted that the

stress in rail under the wheels of this locomotive increases from 8 000 lb. per sq. in. at 5 miles per hour to 9 400 lb. per sq. in. at 60 miles per hour, the increase in stress being small and the stress at 60 miles per hour being much less than that found under the other locomotives.

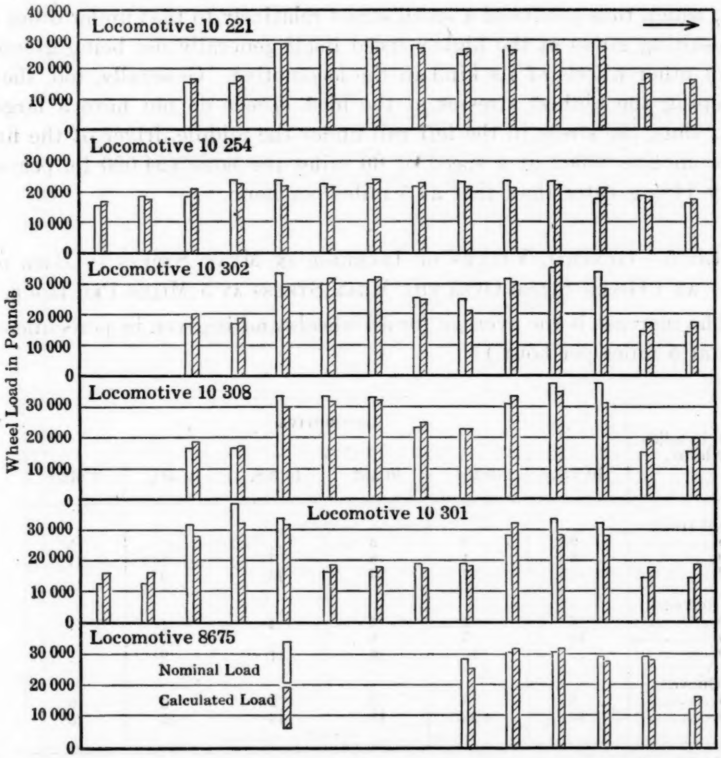


FIG. 13.—REPORTED NOMINAL WHEEL LOADS AND CALCULATED WHEEL LOADS FOR SIX LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

The values already given are averages for the two rails. For the straight track the increase in stress at 60 miles per hour with Locomotive 10254 is greater in the right rail than in the left; for the other electric locomotives it is greater in the left rail, a statement that applies also to the Mikado type locomotive (8675) at 40 miles per hour. The increase for the rail having the greater change is from one-fifth to two-fifths more than the average percentage increase for the two rails. It is evident that the conditions of position and support for the two rails at the test section were such as generally to develop greater speed effect in the left rail than in the right. It is also true that the stresses in the right rail with Locomotive 10254 running at 5 miles per hour are less than those in the left rail (7 500 lb. per sq. in. in the right rail and 8 500 lb. per sq. in. in the left rail), and the stresses in the two rails at 60 miles per hour are nearly the same (9 300 and 9 500 lb. per sq. in., respectively). The records for the several instruments on the same rail vary

also one from another, showing that the effect of speed differs from point to point along the rail.

It was found that although the effect of speed on straight track differed considerably for different wheels, the larger percentages of increase were mostly on wheels having a relatively small load, as, for example, the wheel of a leading truck, which thus produced a small stress relatively to that under other wheels, the resulting stress at the higher speed itself generally not being greater than that of other wheels of its kind in the locomotive. Generally, too, the wheels developing the highest stresses at the high speeds do not have a large speed effect; thus, the stress in the left rail under the middle driver of the first unit of Locomotive 10302 at a speed of 60 miles per hour (24 000 lb. per sq. in.), is only 14% greater than that at 5 miles per hour.

TABLE 5.—General Values of Increase in Mean Stress in Base of Rail at a Given Speed Over the Mean Stress at 5 Miles Per Hour.

(The increase is the average for all wheels and is given in percentage of the stress at 5 miles per hour.)

Speed, in miles per hour.	Locomotive:						Cars.
	10 221.	10 254.	10 302.	10 308.	10 301.	8 675.†	
Straight track :							
25..........	2	− 3	3	..	....	5	..
40..........	6	1	8	4	− 1	11	13
60..........	..	17	24	13	9	..	..
6° Curve :							
30..........	4	− 1	3	4	− 1	..	..
40..........	14	5	6	..	....	..	14
50..........	..	10	10	11	9	..	..
10° Curve :							
25..........	3	8	3	2	6	9	17
40..........	10	18	13	13	17	15*	26*

* At 35 miles per hour.

† The values do not include effect of counterbalance of locomotive drivers. The fourth driver showed on straight track an increase in stress due to counterbalance of 35% for the speed of 40 miles per hour, making with the 11% increase due to speed a total increase of 46 per cent.

The increase in stress with speed found with the Mikado type locomotive was smaller than that observed in previous tests. The speed effect found on straight track with the loaded cars was smaller than expected—14% increase in stress from 5 to 40 miles per hour.

Table 5 also gives for the curved track the average increase in stresses in rail with change in speed, averages for all the wheels of the locomotives or cars being used. As in curved track, centrifugal force, super-elevation of outer rail, turning forces, and restraint of truck frames cause stresses to vary in the inner and outer rails under the several wheels at the different speeds, the values of the ratios given in Table 5 were based on the average of the mean stresses in the two rails under all the wheels of the electric locomotives, under the drivers of the Mikado type locomotive, and under

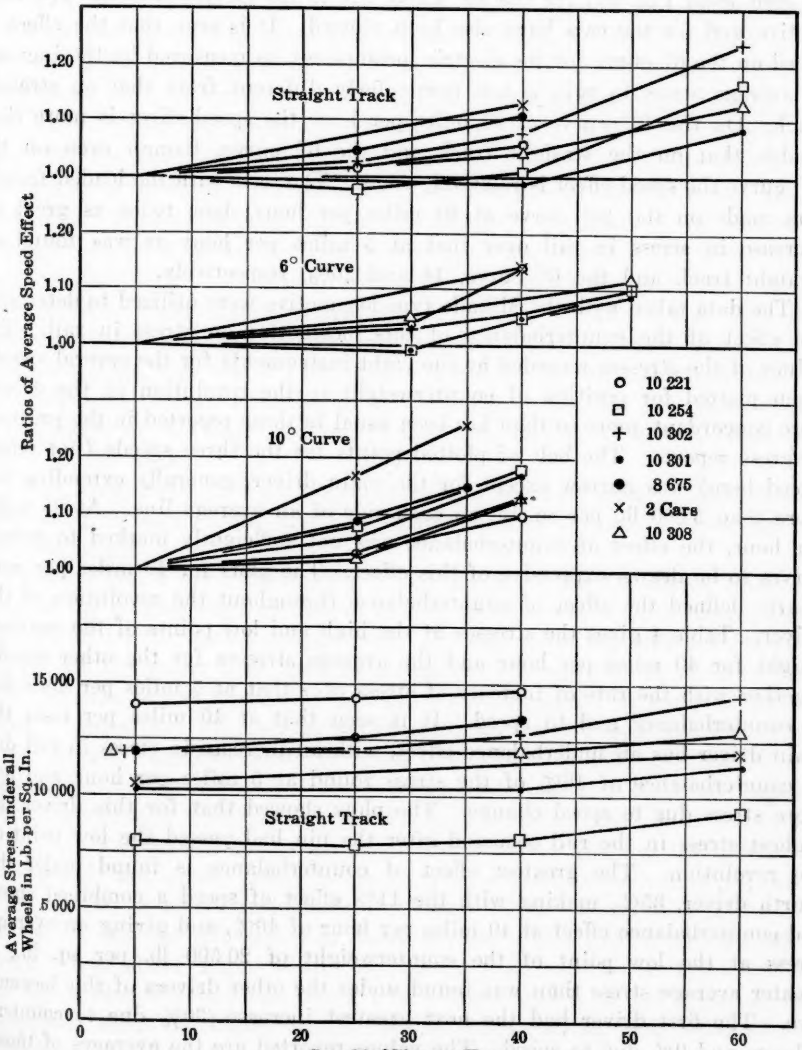


FIG. 14.—RATE OF INCREASE IN STRESS IN RAIL ON STRAIGHT TRACK AND CURVED TRACK IN TERMS OF THE STRESS AT 5 MILES PER HOUR; ALSO VALUES OF STRESS ON STRAIGHT TRACK.

all the wheels of the loaded freight cars. This method is not an entirely rational one, but it may be expected to give a fairly good way of judging of the effect of speed on curved track. The general trend of the effect of speed with the electric locomotives on the 6° curve and the 10° curve may be seen from Fig. 14; for the 10° curve the ratios for the Mikado type locomotive and for the cars have also been plotted. It is seen that the effect of speed on the 6° curve for the electric locomotives, as measured by the increase in average stress in rail, is not particularly different from that on straight track. On the 10° curve at 40 miles per hour the speed effect is more than double that on the straight track and the 6° curve, though even on the 10° curve the speed effect is relatively small. The tests with the loaded freight cars made on the 10° curve at 40 miles per hour show twice as great an increase in stress in rail over that at 5 miles per hour as was found on straight track and the 6° curve, 14 and 26%, respectively.

The data taken with the Mikado type locomotive were utilized to determine the effect of the counterbalance of this locomotive on stress in rail. The values of the stresses recorded by the eight instruments for the several speeds, when plotted for position of counterweight in the revolution of the driver, were concordant, more so than has been usual in those reported in the previous progress reports. The belt of plotted points for the three speeds (not reproduced here) was narrow except for the main driver, generally extending not more than 5 000 lb. per sq. in. on each side of an average line. At 25 miles per hour, the effect of counterbalance was not sufficiently marked to permit curves to be drawn expressive of this effect. The plots for 40 miles per hour clearly defined the effect of counterbalance throughout the revolution of the driver. Table 4 gives the stresses at the high and low points of the counterweight for 40 miles per hour and the average stresses for the other speeds, together with the rate of increase of stress over that at 5 miles per hour due to counterbalance and to speed. It is seen that at 40 miles per hour the main driver has an underbalance effect, with an increase in stress in rail due to counterbalance of 16% of the stress found at 5 miles per hour and 8% more stress due to speed change. The plots showed that for this driver the highest stress in the rail occurred after the pin had passed the low point of the revolution. The greatest effect of counterbalance is found under the fourth driver, 35%, making with the 11% effect of speed a combined speed and counterbalance effect at 40 miles per hour of 46%, and giving an average stress at the low point of the counterweight of 20 500 lb. per sq. in., a greater average stress than was found under the other drivers of this locomotive. The first driver had the next greatest increase, 30% due to counterbalance and 9% due to speed. The values reported are the averages of those found in the two rails and for the eight instruments used. Except perhaps for the fourth driver the counterbalance effect of this locomotive compares favorably with that of most of the Mikado type locomotives heretofore used, and the speed effect is markedly low for this type of locomotive.

9.—Lateral Bending Stresses and Lateral Movement of Rail on Straight Track.—As noted in the previous progress reports, the rail of straight track

bends laterally, sometimes outwardly and sometimes inwardly, as the wheels pass over it. The differences in the stresses in the two edges of the base of rail may be taken to measure the intensity of the lateral flexure, though it does not give the amount of the lateral movement or even its direction. The ratio of the stress at the outside edge of the base of rail to the mean of the stresses at the two edges is a convenient ratio to use in making comparisons of the lateral bending effect at different points along the track or on different runs or under different wheels, if the load of the wheel or the mean stress under the wheel do not differ greatly. If the ratio is greater than 1.00, the rail has been bent outwardly of the track; if less than 1.00, the bending is inwardly. Unless there is considerable tilting of the rail under the load in an unusual way, outward bending of the head of the rail goes with outward bending at the base, a dissimilar movement would imply very great twisting action. It will be well to keep in mind that a ratio of stress at the outside edge to mean stress having a value of 1.20 means that the stress at the outside edge is 1.50 times that at the inside edge, one of 1.33, that it is double, and one of 1.50, that it is treble. As the amount, direction, and conditions of action in the lateral bending of the rail may throw light on the action of the track and its maintenance, it will be well to study the results of the tests.

It has been found that the lateral bending of the rail differs with different wheels and different runs and also that it may vary at different points along the track.

Tables 6 and 7 give for straight track, for the different speeds, ratios of stress at the outside edge of base of rail to mean stress for the several locomotives and for the loaded freight cars. The values are the averages for each rail for all the runs; they thus represent the average results for the four points on each rail at which the instruments were placed. A study of Tables 6 and 7 shows that the average ratios are both greater and less than 1.00, that those less than 1.00 rarely go below 0.90, and that although most of the averages for the several speeds are less than 1.10, yet a number are greater than 1.20, and one wheel at one speed averages above 1.30. It is seen that the average for the left rail is less than that for the right, but the difference in general is not great. Most locomotives have fairly even values for most of the wheels, as note Locomotives 10254 and 10308. Locomotive 10302 has two drivers on the left side and three drivers and a trailer on the other giving values above 1.10. With the Mikado type locomotive, three drivers and two tender wheels range above 1.10. Three wheels of the eight on the loaded cars show the higher ratios. An examination has been made of the contours of the tires of the locomotives, but no definite correlation was found between the condition of the tires and observed stress ratios.

As a relatively few applications of the lateral pressures that give the extreme values (high or low) may be expected to affect track maintenance more seriously than many applications near the average value of the ratio, it will be well to consider the range and distribution of individual values. Fig. 15 shows individual observations for the Mikado type locomotive, the

TABLE 6.—VALUES OF THE AVERAGE RATIO OF THE STRESS AT THE OUTSIDE EDGE TO MEAN STRESS IN BASE OF RAIL OF STRAIGHT TRACK FOR THE LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

Locomotive.	Speed, in miles per hour.	Rail.	Rear End Truck. 2.	Rear End Truck. 1.	Driver. 8.	Driver. 7.	Driver. 6.	Driver. 5.	Driver. 4.	Driver. 3.	Driver. 2.	Driver. 1.	Head End Truck. 2.	Head End Truck. 1.
10221	5	Left......	0.94	1.05	0.97	1.06	1.04	0.98	1.00	1.01	1.01	1.01	0.94	1.13
		Right....	1.05	1.04	1.05	1.04	1.07	1.05	1.03	1.05	1.02	1.02	1.03	1.16
	25	Left......	0.99	1.07	0.97	1.01	1.04	0.96	0.99	0.99	1.03	0.97	1.04	1.12
		Right....	1.02	1.05	1.05	1.04	1.10	1.06	1.05	1.06	1.05	1.03	1.06	1.11
	40	Left......	0.96	1.14	0.97	1.05	1.13	1.00	1.03	1.04	1.05	1.02	0.94	1.14
		Right....	1.01	1.04	1.04	1.04	1.16	1.05	1.01	1.04	1.02	0.99	1.00	1.08

Locomotive.	Speed, in miles per hour.	Rail.	Rear End Truck. 2.	Rear End Truck. 1.	Driver. 12.	Driver. 11.	Driver. 10.	Driver. 9.	Driver. 8.	Driver. 7.	Driver. 6.	Driver. 5.	Driver. 4.	Driver. 3.	Driver. 2.	Driver. 1.	Head End Truck. 2.	Head End Truck. 1.
10254	5	Left......	....	0.94	0.96	0.97	1.01	0.92	0.94	0.95	0.93	1.01	0.98	1.13	0.97	1.02	....	0.99
		Right....	....	0.96	0.93	1.01	1.03	0.92	0.99	1.03	1.01	0.93	0.89	1.16	0.90	0.98	....	1.01
	25	Left......	....	0.98	0.96	0.98	0.99	0.94	0.95	0.95	1.10	1.05	1.00	1.17	0.98	1.11	....	0.98
		Right....	....	0.95	0.96	1.01	1.04	0.99	1.08	1.10	1.07	1.00	1.01	1.23	0.96	1.05	....	1.03
	40	Left......	....	1.06	1.11	1.03	1.11	1.02	1.11	1.01	1.05	1.14	1.15	1.23	1.03	1.13	....	0.99
		Right....	....	1.02	1.11	1.07	1.05	1.05	1.05	0.98	1.01	1.06	0.95	1.26	0.99	1.07	....	1.02
	60	Left......	....	1.06	1.01	0.95	1.07	1.10	1.11	1.15	1.05	1.22	1.23	1.25	1.11	1.17	....	1.08
		Right....	....	1.01	0.96	1.02	1.06	1.09	1.06	1.26	1.06	1.07	1.02	1.18	0.99	0.95	....	1.03

Locomotive.	Speed, in miles per hour.	Rail.	Rear End Truck. 2.	Rear End Truck. 1.	Tender. 4.	Tender. 3.	Tender. 2.	Tender. 1.	Trailer. 1.	Driver. 4.	Driver. 3.	Driver. 2.	Driver. 1.	Head End Truck. 2.	Head End Truck. 1.
8675	5	Left......	....	....	0.95	1.04	0.99	1.16	0.84	1.06	0.98	1.08	1.12	....	1.04
		Right....	....	....	0.95	0.97	0.91	1.17	1.07	1.15	1.08	1.15	1.12	....	0.99
	25	Left......	....	....	0.92	1.01	1.02	1.16	0.82	1.07	0.92	1.03	1.09	....	0.98
		Right....	....	....	0.95	1.02	0.91	1.19	1.04	1.18	1.15	1.17	1.04	....	0.99
	40	Left......	....	....	0.93	1.03	1.05	1.14	0.84	1.04	0.94	1.08	1.12	....	1.02
		Right....	....	....	0.93	0.97	0.92	1.10	1.03	1.15	1.05	1.16	1.02	....	1.01

TABLE 7.—VALUES OF THE AVERAGE RATIO OF THE STRESS AT THE OUTSIDE EDGE TO MEAN STRESS IN BASE OF RAIL OF STRAIGHT TRACK FOR THE LOCOMOTIVES AND CARS OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

Locomotive.	Speed, in miles per hour.	Rail.	Rear End Truck. 2.	Rear End Truck. 1.	Driver. 6.	Driver. 5.	Driver. 4.	Trailer. 4.	Trailer. 3.	Trailer. 2.	Trailer. 1.	Driver. 3.	Driver. 2.	Driver. 1.	Head End Truck. 2.	Head End Truck. 1.
10 302	5	Left	0.88	0.99	0.90	0.98	1.19	....	....	0.85	1.01	1.10	1.13	0.98	1.06	1.03
		Right	0.97	.87	1.00	1.01	1.06	....	....	1.08	1.16	1.22	1.22	1.33	0.73	1.01
	25	Left	0.93	0.95	0.90	0.92	1.06	....	....	0.80	0.95	1.05	1.13	0.95	1.03	0.99
		Right	0.98	0.82	0.97	1.00	1.11	....	....	1.04	1.11	1.23	1.23	1.28	0.89	1.00
	40	Left	0.92	0.94	0.94	0.98	1.09	....	....	0.89	1.02	1.09	1.15	0.97	1.09	1.04
		Right	1.05	1.00	1.01	0.99	1.10	....	....	1.03	1.11	1.21	1.24	1.27	0.76	1.02
	60	Left	1.02	1.00	0.97	0.99	1.10	....	....	0.89	1.06	1.05	1.13	1.05	1.08	1.00
		Right	1.04	1.00	1.01	1.01	1.06	....	....	1.03	1.05	1.12	1.19	1.21	0.99	1.07
10 308	5	Left	0.97	1.05	1.09	1.01	1.05	....	....	1.01	1.02	1.08	0.97	0.96	1.06	1.05
		Right	1.06	0.97	1.15	1.09	1.10	....	....	1.02	1.07	1.05	0.97	1.01	0.93	1.02
	40	Left	0.98	0.92	1.16	1.08	1.01	....	....	1.05	1.03	1.07	0.98	1.02	1.06	1.08
		Right	1.01	1.09	1.24	1.10	1.08	....	....	1.02	1.07	1.06	1.01	1.01	1.14	1.05
	60	Left	0.98	0.95	1.18	1.06	1.08	....	....	1.03	1.05	1.07	1.04	1.07	1.04	1.02
		Right	1.00	1.08	1.21	1.05	0.97	....	....	1.02	1.03	1.05	0.93	1.04	1.13	1.06
10 801	5	Left	0.98	0.95	0.97	0.94	0.96	1.04	1.02	1.07	1.06	0.99	0.99	1.01	1.20	0.92
		Right	1.05	1.10	1.04	0.99	1.05	1.20	1.09	1.08	0.96	1.07	1.04	1.06	1.03	1.21
	40	Left	0.95	0.93	1.00	0.93	0.94	1.05	0.94	0.98	0.97	0.95	0.97	0.98	1.13	0.99
		Right	1.12	1.21	1.06	0.97	1.00	1.14	1.09	1.12	1.09	1.05	1.04	1.01	0.94	1.21
	60	Left	0.99	1.22	1.10	1.04	1.08	1.11	1.08	0.92	1.15	1.07	1.02	1.14	1.37	1.05
		Right	0.94	1.17	0.97	0.93	0.96	1.10	1.06	1.06	1.03	1.01	1.00	0.97	0.97	1.18

			Second Car. 4.	Second Car. 3.	Second Car. 2.	Second Car. 1.	First Car. 4.	First Car. 3.	First Car. 2.	First Car. 1.
Loaded cars............	5	Left	0.94	1.08	1.00	1.00	0.95	1.06	1.05	1.08
		Right	0.99	1.13	0.94	0.99	0.91	1.10	0.95	1.23
	40	Left	0.99	1.07	1.06	1.08	1.06	1.08	1.11	1.04
		Right	1.03	1.11	1.02	1.02	1.05	1.18	1.10	1.06

values for the several instruments being distinguished by symbols and each instrument being assigned a part of the space given for a wheel and in the order of the instruments along the track. It is seen that for the Mikado type locomotive the range is from less than 0.60 to more than 1.40. It is also seen that although the average ratio on the two rails for all the drivers and the trailer is 1.07, there are a large number of individual values between

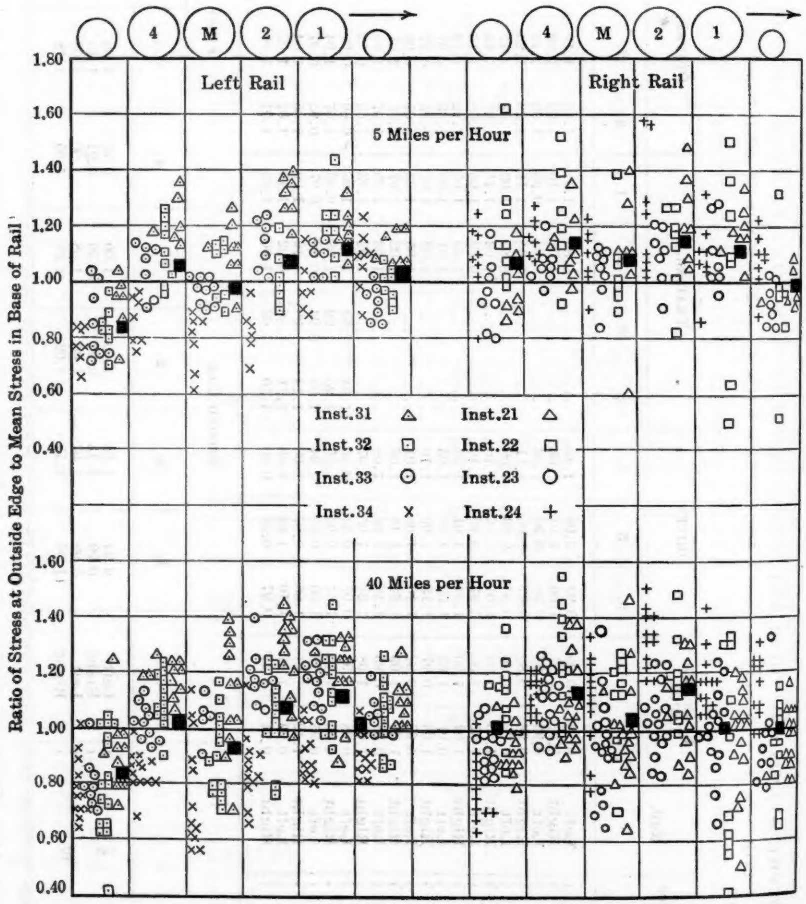


FIG. 15.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOCOMOTIVE 8675 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

1.20 and 1.50, and relatively few less than 0.80. It is apparent that many relatively large outward pushes on the rail were given in the passage of the drivers over this track and that the effect of the outward bending of the rail upon upkeep would be larger than the indication given by the average of all the ratios. It is noticeable that at a particular instrument one wheel would give a relatively high average value and another wheel a lower one, and at another instrument the reverse was true. In general, the range of values

recorded by an instrument was considerable, the amount of the range being much the same for all the instruments, but, in a few cases, at a single wheel the values were quite compactly grouped.

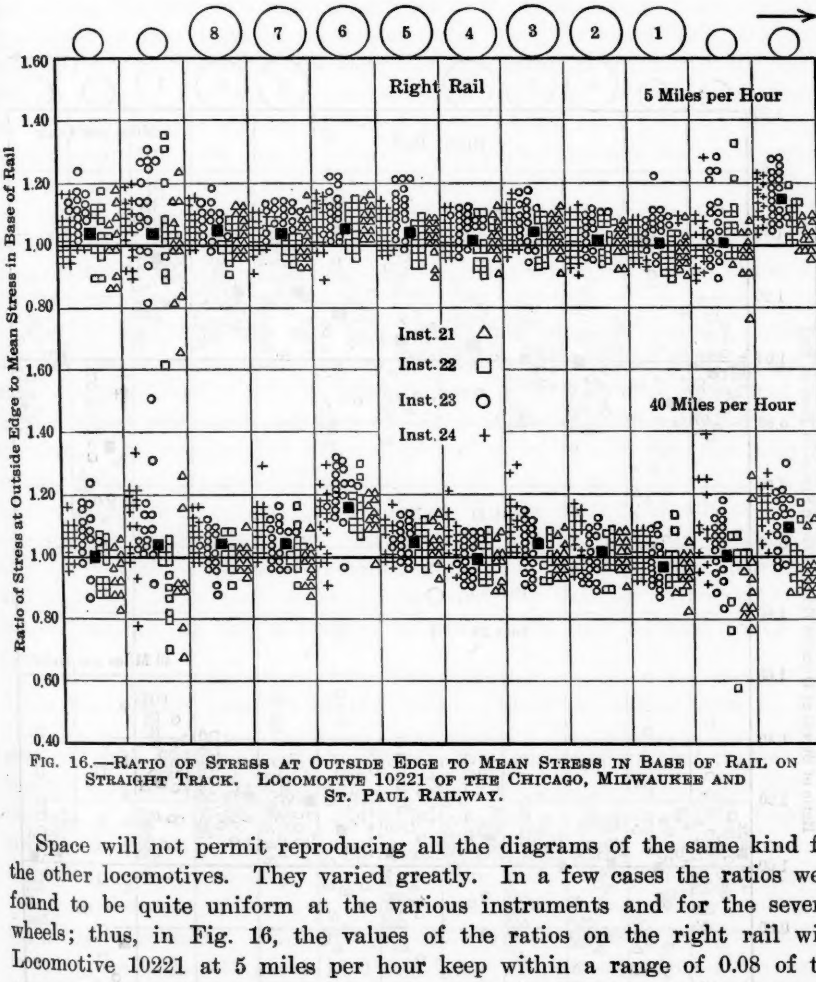


FIG. 16.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOCOMOTIVE 10221 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

Space will not permit reproducing all the diagrams of the same kind for the other locomotives. They varied greatly. In a few cases the ratios were found to be quite uniform at the various instruments and for the several wheels; thus, in Fig. 16, the values of the ratios on the right rail with Locomotive 10221 at 5 miles per hour keep within a range of 0.08 of the average ratio and are very compactly grouped. More generally, however, the values of the ratios are widely distributed and may vary for different wheels; thus, Locomotive 10302 at 40 miles per hour (see Fig. 17) gives ratios of stresses in the right rail that vary from 0.40 to 1.55, this range occurring under adjoining wheels. It is evident that the variations in values (high and low) at the several instruments on one rail differed considerably from those at the several instruments on the other rail. The average value for the two rails also differed. In some cases, as for one instrument near a rail joint, the values were in the high part of the range (or the low part) for all the locomotives, this being the first instance in the test work that the

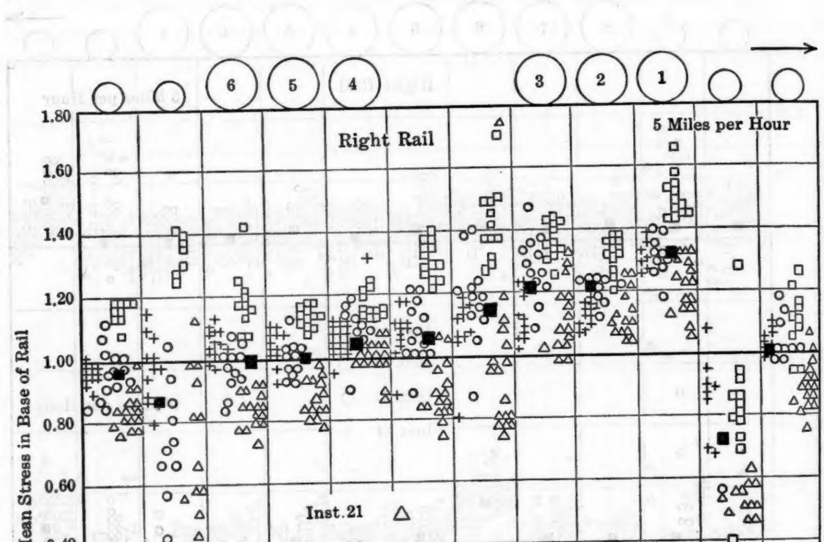


FIG. 17.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. LOCOMOTIVE 10302 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

ratios for a given instrument near a joint were consistently higher or lower than those at the other side of the joint. Generally, however, a given wheel develops at a given instrument little variation in ratios, the range from an average usually being not more than 5 per cent. Some wheels differ markedly from most of the others of the locomotives; thus, the third right driver of Locomotive 10254 for all speeds generally gave ratios at the four instruments between 1.10 and 1.60, the average being 1.21, while for the remaining wheels of the right side the average ratio was slightly more than 1.00 and the individual observations are fairly uniformly distributed through the usual range; on the left rail the range in ratios is still greater and for certain wheels there are marked differences in the values at different instruments. It must be borne in mind that the stresses developed by Locomotive 10254 are small as compared with the other locomotives and that given variations in stress will show a higher range in the ratios.

For the loaded freight cars (for which fewer observations are available than for the locomotives) the values of the ratio of stress at outside edge to mean stress in base of rail, in general, have a much smaller range and are more compactly grouped and vary less from instrument to instrument along the track. At 5 miles per hour, only one wheel of the first car gave an average value of the ratio of more than 1.20 and, at 40 miles per hour, the higher values are relatively few in number. In the second car only one wheel gave a ratio greater than 1.10. Strangely, at 5 miles per hour, most of the high individual ratios were recorded under one wheel by one instrument and at 40 miles per hour under another wheel by another instrument. The lateral pressure on the rail given by these cars may be said to be not excessive.

As the section modulus of the rail, $\frac{I}{c}$, about a vertical axis is only about one-fifth of that about a horizontal axis, it is apparent that the bending moment corresponding to a given lateral bending stress is only one-fifth of the vertical bending moment which would give an equal vertical bending stress. The value of the observed lateral bending stress may be found by multiplying a mean stress in base of rail by a factor obtained by subtracting 1.00 from the ratio of stress at outside edge to mean stress corresponding to the given observation. Thus, for the right first driver of Locomotive 10302, by using the ratio 1.33 given in Table 7 and the corresponding mean stress in base of rail given in Table 3, 13 500 lb. per sq. in., the lateral bending stress is seen to be 4 500 lb. per sq. in. An individual observation giving a ratio of 1.38 with a mean stress of 22 000 lb. per sq. in. corresponds to a lateral bending stress of 8 400 lb. per sq. in.

It is evident from a study of the tests that the lateral bending of the rail is caused both by the condition of the rail and the rail support (tie and underlying ballast) and by the action of the locomotive and cars. At two of the eight instruments, there was marked outward bending of the rail for all the locomotives, as measured by the average stresses for all wheels, and at one instrument marked inward bending. At all the other points of

attachment of instruments, the several locomotives gave different effects. That is, at three of the eight points the support of the rail was such as to favor the lateral bending of the rail one way or the other. At the other five points the locomotive and cars were evidently the governing cause of the lateral bending. It is apparent, of course, that the variability in the lateral bending shown by the considerable range in bending stresses and ratios is due to variations in the action of the locomotive in passing over the track on different runs.

Two trial runs at slow speed with Locomotive 10302 gave maximum outward lateral movements of the head of one rail of 0.03 and 0.05 in. when the wheels were passing over the point of observation. It is seen that the lateral movement is slight in this case and that nothing of value was learned from the few measurements made.

To learn something of the position of the flanges and treads of the wheels with respect to the head of the rail a few tests on straight track were made near the end of the work by taking impressions on copper wire as the locomotive passed over the test section. As the method of using a punch mark on the head of the rail had not then been developed, the position of the wheel bearing on the rail is uncertain except for the wheels having flange contact with the side of the head. It was found for Locomotive 8675 that certain wheels ran close to the gauge side of the rail at all speeds. Some wheels showed a broad contact with the rail, others a more concentrated contact. The flange of the first driver on the right side and usually also that of the fourth driver, ran close to the rail. The flange of the trailer and first wheel of the tender on the left side also ran close to the rail. In the runs at 40 miles per hour with Locomotive 10308, the flanges of the drivers on the right side of the second unit ran close to the rail. Under the other wheels the flanges were away from the rail and the pressures were concentrated near the center of the head of the rail or somewhat inside it. At 40 miles per hour, the flange of the first wheel on the left side ran close to the rail on one run and that on the other side on the other run. The flanges of other wheels also ran close to the rail. These observations are described at this length in order to bring out the variability of the positions and directions of pressures of wheels on rail and thus to point to some of the variable conditions that result in great variations in the amount and character of both vertical and lateral bending stresses in the rail.

10.—The Action of Curved Track.—In the third progress report, under the heading, "The Action of Curved Track",* the way in which the passage of a locomotive and cars around curved track may be expected to affect the track through transfer of load from rail to rail and wheel to wheel and the way in which lateral pressures on the rail may be produced, were discussed. Centrifugal force, transverse inclination of track, flange and tread lateral pressures, spreading of rails, and other matters were considered. In particular, it was brought out that the lateral force of the flange of the leading truck wheel or first driver or both against the outer rail necessary to change

* *Transactions*, Am. Soc. C. E., Vol. LXXXVI (1923), p. 973.

the direction of all the drivers grouped in a single frame in traversing a curve brought into action a lateral pressure against the inner rail by the wheels at or near the center of rotation about which the turning of the locomotive takes place and likewise an outward pressure on the outer rail by the rear outside driver. It was further shown that the vertical load transferred to the rail by the inside driver nearest the center of rotation might be much greater than its normal vertical load. The lateral forces thus set up and the increase in the load on one or more of the drivers which may be developed, greatly increase the lateral and vertical moments developed in the rail and therefore have important effects on track maintenance as well as on the rail itself. For details of the discussion reference is made to the third progress report, but a résumé of some of the statements and conclusions follows:

(*a*).—The longitudinal slip along a rail of the inside or outside wheel of a pair on the same axle made necessary by the difference in length of the two rails of curved track is considered to affect the lateral forces acting on the rails only as it may modify the coefficient of friction and thus also the magnitude of the force required to produce lateral slipping of the wheel on the rail. The effect of the coning of the wheels is considered to be negligible as an element in the problem.

(*b*).—The centrifugal force and the transverse inclination of the track necessarily affect the vertical and lateral pressures on the rails to an appreciable extent. The following approximate formulas were given for the pressure on the outer and the inner rail, respectively, to be expected from analytical considerations to correspond to the weight, W, applied to the rail through one wheel for straight and level track, when the speed is such that the centrifugal force is negligible:

$$R' = W\left(1 - 2\frac{h}{g}\frac{e}{g}\right) \ldots\ldots\ldots\ldots\ldots\ldots (29)^*$$

$$R'' = W\left(1 + 2\frac{h}{g}\frac{e}{g}\right) \ldots\ldots\ldots\ldots\ldots\ldots (30)^*$$

In these equations, g is the distance between the bearing points of the wheels on the two rails (taken in calculations as 59 in. for the curved track used in the tests), e is the super-elevation of the outer rail, and h is the distance of the center of gravity of the locomotive or car from the level of the top of the rails. The equations were based on the assumption that the bodies of the locomotive and cars retain the same positions with respect to the axles of the wheels as are found on straight track. The tilting due to changed compression of the springs may modify the pressures somewhat.

Taking the load on a rail as equal to the algebraic sum of that due to Equation (29) or Equation (30) and that due to the centrifugal force, approximate formulas for the reactions of the outer and inner rails, respectively, were given as:

$$R_1 = W\left[\left(1 - 2\frac{e}{g}\frac{h}{g}\right) + \frac{2V^2D}{85\,800}\left(\frac{h}{g} + \frac{1}{2}\frac{e}{g}\right)\right] \ldots\ldots (31)^\dagger$$

* *Transactions*, Am. Soc. C. E., Vol. LXXXVI (1923), p. 979.

† *Loc. cit.*, p. 980.

$$R_2 = W\left[\left(1 + 2\frac{e}{g}\,\frac{h}{g}\right) - \frac{2\,V^2 D}{85\,800}\left(\frac{h}{g} - \frac{1}{2}\,\frac{e}{g}\right)\right] \ldots\ldots (32)^*$$

in which V is the speed, in miles per hour, and D is the degree of curve. Likewise, the sum of the resulting lateral reaction or pressure against the outer rail of the curved track, F_1, and that against the inner rail, F_2, is obtained approximately by the equation:

$$F_1 + F_2 = \frac{2\,W\,V^2 D}{85\,800} - 2\,W\frac{e}{g} \ldots\ldots\ldots\ldots (33)^*$$

the division of the lateral force between F_1 and F_2 being indeterminate.

(*c*).—In changing the direction of a group of drivers, as the locomotive traverses a curve, the outer rail presses against the flange of the outer front driver, or the outer wheels of the front truck, or both, according to conditions of speed, length of wheel-base, flexibility of frame and connections with other trucks, and other matters. This outer front driver and the inner driver on the same axle must be made to slip laterally on the rails inwardly of the curve. At some driver farther back, generally at a driver on the inner rail, there will be a center of rotation of the group of wheels connected to a single frame, about which the turning action of the frame takes place. The drivers ahead of the center of rotation, except the leading outer driver when it participates in the turning or directing action, will give pressures inwardly of the curve on both inner and outer rails. The drivers behind the center of rotation will swing outwardly of the curve and slip outwardly, and thus pressure in this direction will be developed against both the outer and the inner rail. In Fig. 18 the direction of the pressures of the wheels of a Mikado type locomotive against the rails is indicated. B is the center of rotation.

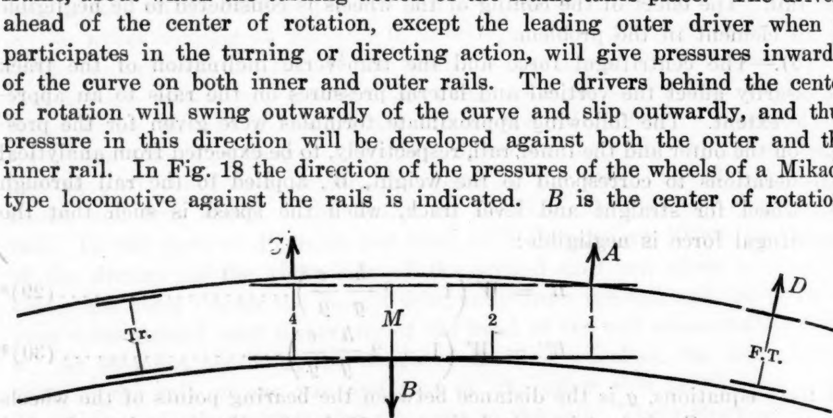


FIG. 18.—DIRECTION OF PRESSURE OF THE WHEELS OF A MIKADO TYPE LOCOMOTIVE ON THE RAILS OF CURVED TRACK.

The turning forces are at D and frequently at A also, flanges of the guiding wheels pressing against the head of the outer rail. It is evident that the lateral pressure of the wheel on the inner rail at the center of rotation, B, may constitute the principal force in opposition to the several lateral forces that are acting in the opposite direction. The pressure developed by the other drivers will be smaller in magnitude and will vary in direction according to conditions. The trailer may give pressures outwardly of the curve or inwardly according to the restraint, the coupling connection, and the degree of the curve. The deflections and lateral movement of the two rails as the locomotive moves around the curve, themselves take a part in fixing the positions of the drivers and thus affect the development of the lateral reactions.

* *Transactions*, Am. Soc. C. E., Vol. LXXXVI (1923), p. 980.

It is seen that the lateral forces tending to bend or to straighten the two rails, and thus to affect the whole track structure and its maintenance, are dependent on numerous factors.

In the four-wheel truck of a car, the turning is generally done by the flange of the outer forward wheel, and the rear wheels of the truck may be expected to approach a position radial to the curve.

11.—Results of Tests on Curved Track.—Space will permit the presentation of only a part of the data, but representative parts have been selected and general average values are given in the tables and diagrams.

Figs. 19 to 30, inclusive, give the stresses at the inside edge and the outside edge of the base of the outer and the inner rail under the wheels of six locomotives and two loaded freight cars at several speeds on the 6° curve and the 10° curve. The values given are the averages of the records of the four instruments on a rail for the runs made, representing generally about sixty observations for each value plotted. The variations of individual values from the average value reported were much the same as on straight track, but the differences between averages for the four instruments on a rail were greater than those for straight track, indicating that variations in the curvature of the rail and the conditions of its support affect the stresses developed to a greater degree than is the case with straight track.

In making comparisons between tests, reference may be made to the properties of the 90-lb. rail section given in Article 4, "The Track", and the weights of the locomotives and cars given in Figs. 1 and 2.

12.—Vertical Loads and Vertical Bending Stresses on Curved Track.—The mean stress in the base of rail (the average of the stresses observed at the two edges and generally called the vertical bending stress) is taken as representative of the bending of the rail in a vertical plane, or, more strictly, of the bending in a direction normal to the inclination of the track, the two effects being practically identical for the super-elevation of track used in the tests, since the cosine of the angle of inclination is very close to unity. Although stresses in rail are not exactly proportional to the loads when the same total load is differently distributed among the several wheels or differently divided between the two rails, yet when the differences are not great the sum of the stresses in the rail under all the wheels will generally differ little on account of different divisions or distribution of load, and summations of the stresses for each rail may be useful in making comparisons and in checking up on the action of the locomotive and track. As has already been stated in Article 7, "Results of Tests on Straight Track", by means of the analytical method given in the first progress report and the solution of unknowns in equations based thereon, the load which will produce the given mean stress may be estimated quite approximately. It will be seen then that the values of the mean stresses in base of rail (vertical bending stresses) not only will permit comparisons with the stresses in straight track to be made, but will enable the vertical load to be distinguished from the lateral forces or loads that are also developed in passing around the curve. A close estimate may then be made of the distribution of vertical load among the wheels and between

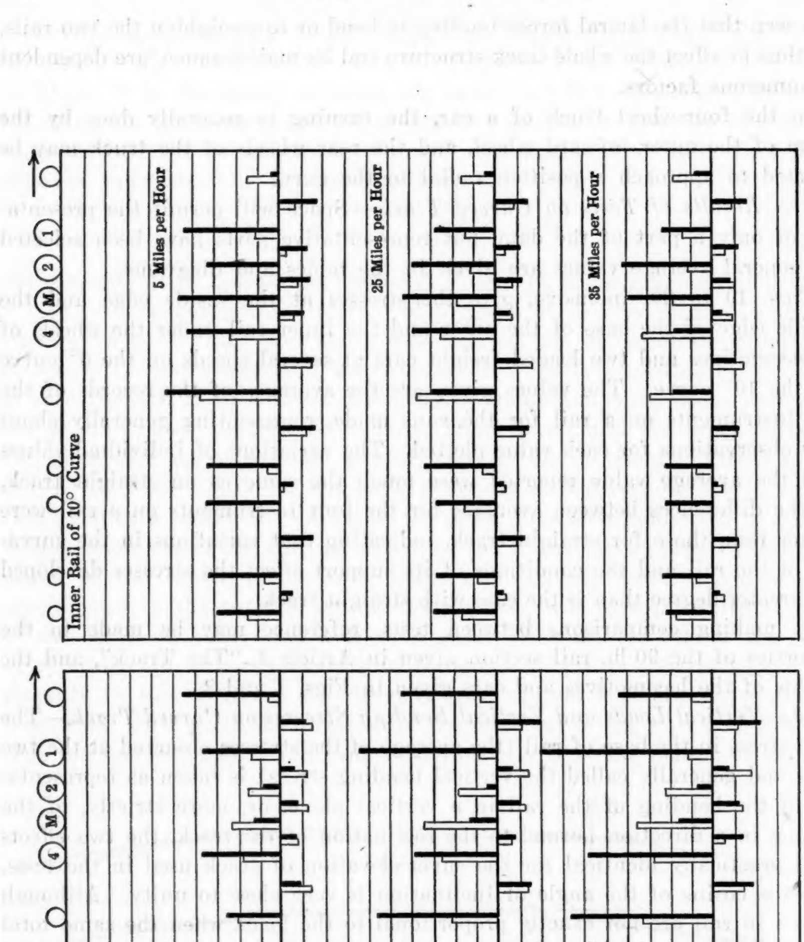


FIG. 19.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 10° CURVE. LOCOMOTIVE 8675 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

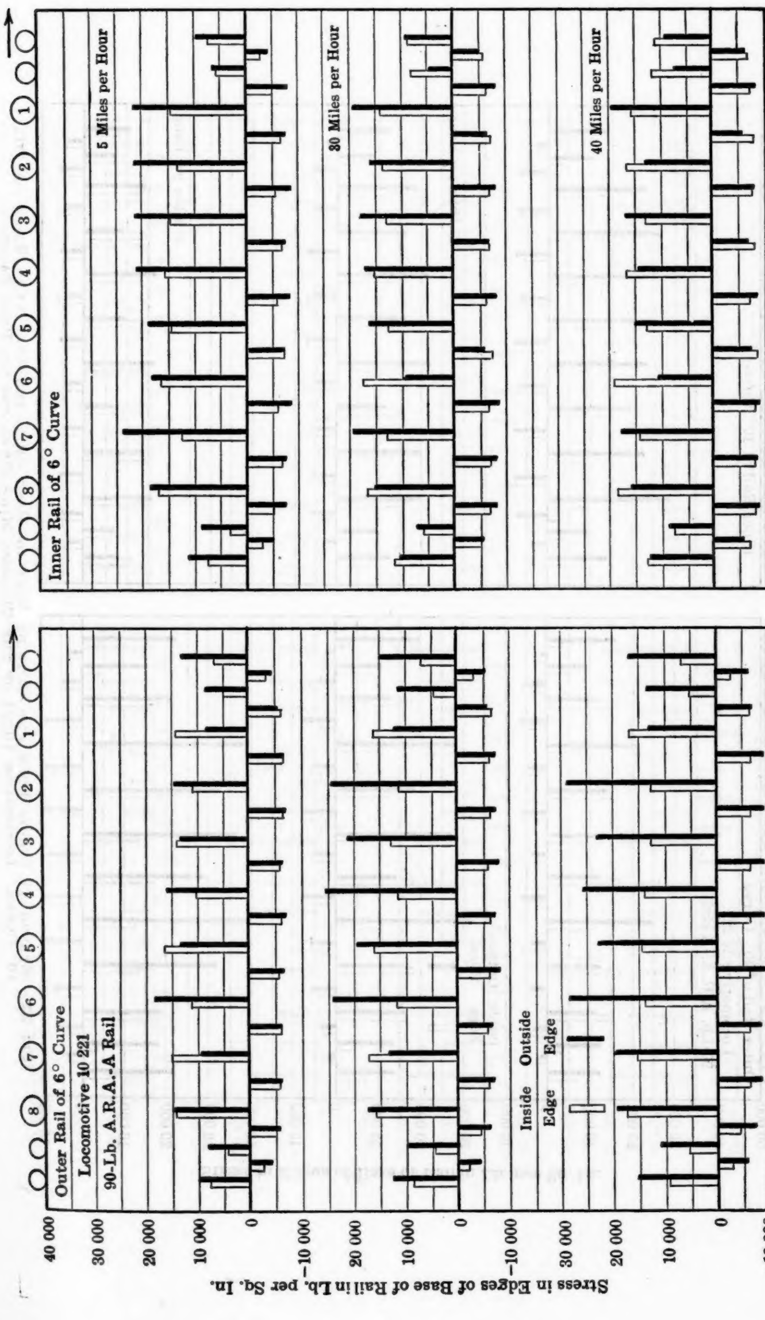


FIG. 20.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 6° CURVE. LOCOMOTIVE 10221 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

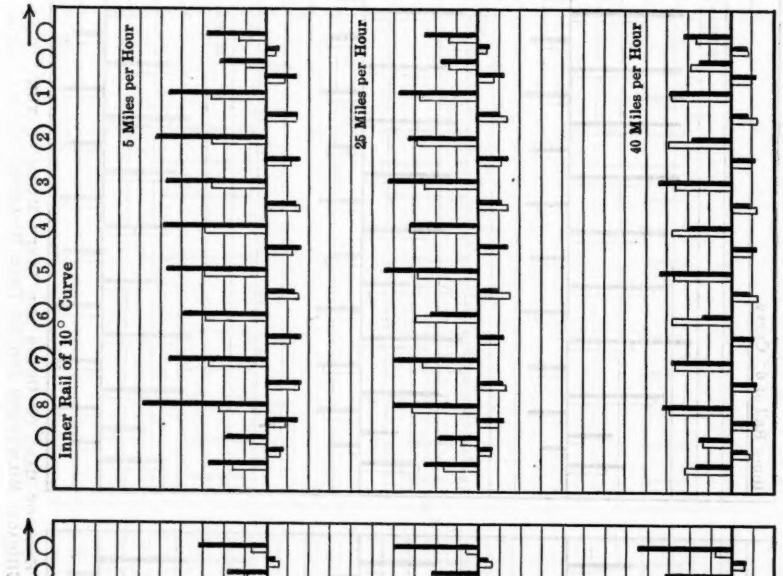


FIG. 21.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 10° CURVE. LOCOMOTIVE 10221 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

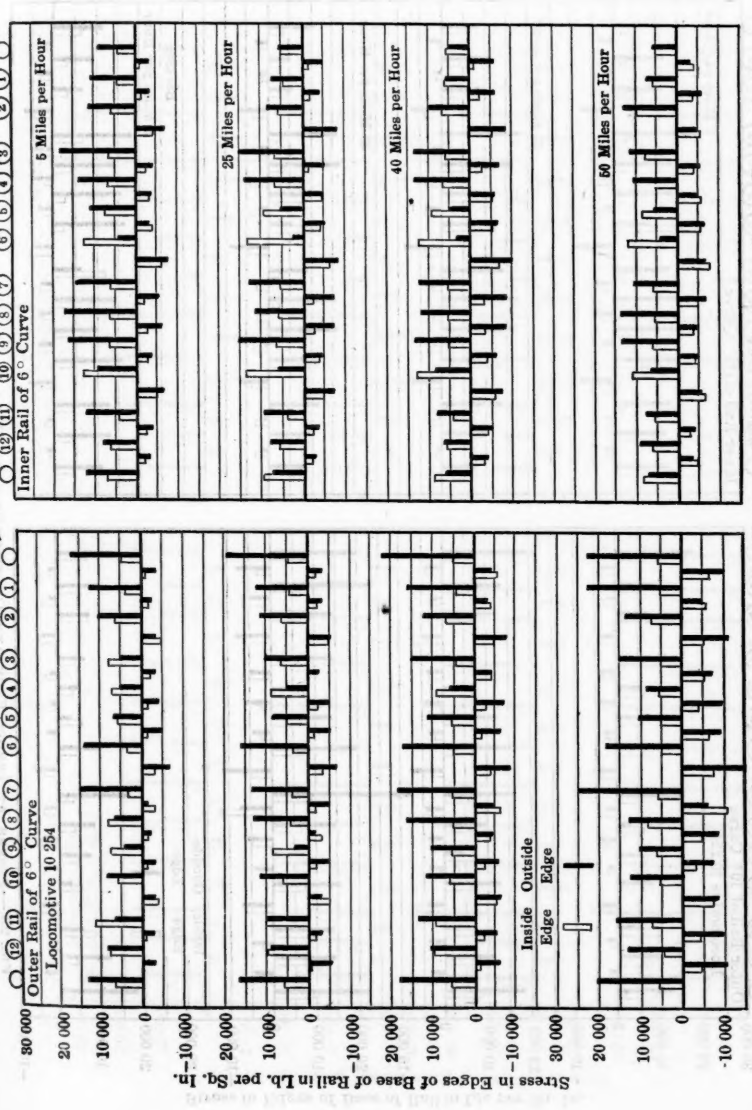


FIG. 22.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 6° CURVE. LOCOMOTIVE 10254 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

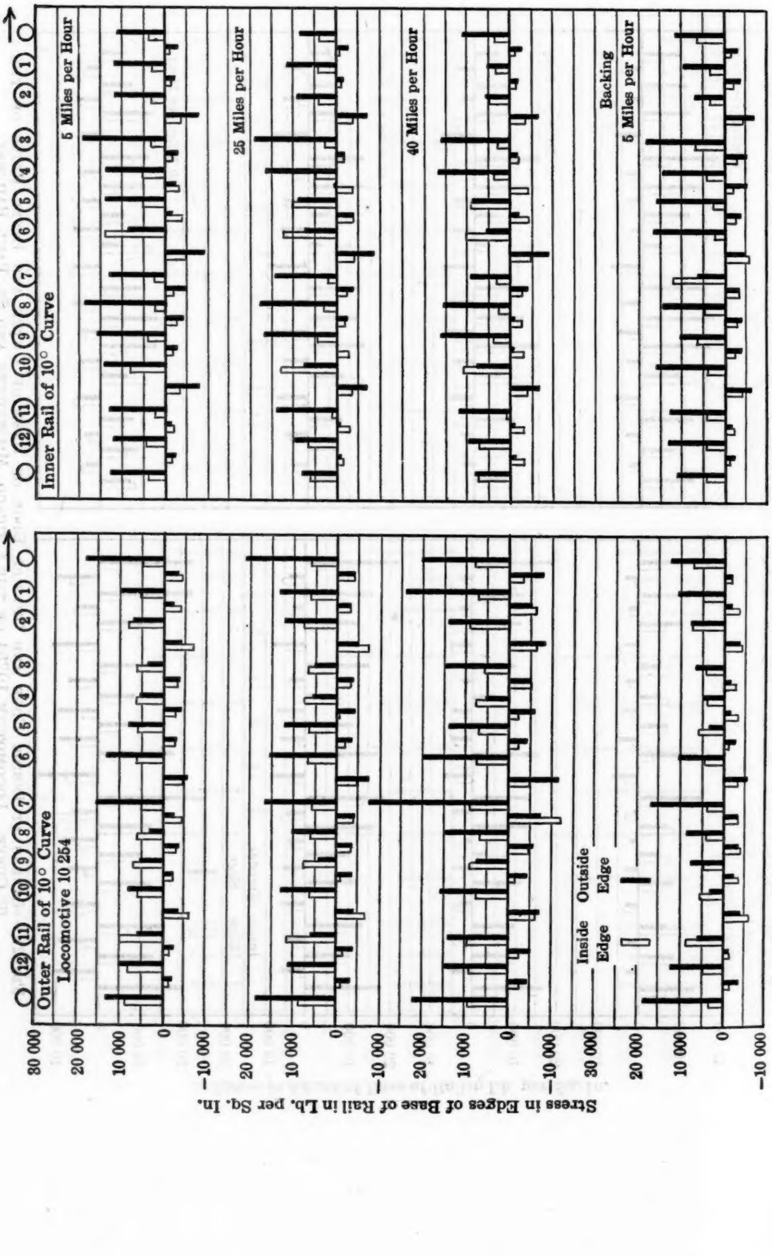


FIG. 23.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 10° CURVE. LOCOMOTIVE 10254 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

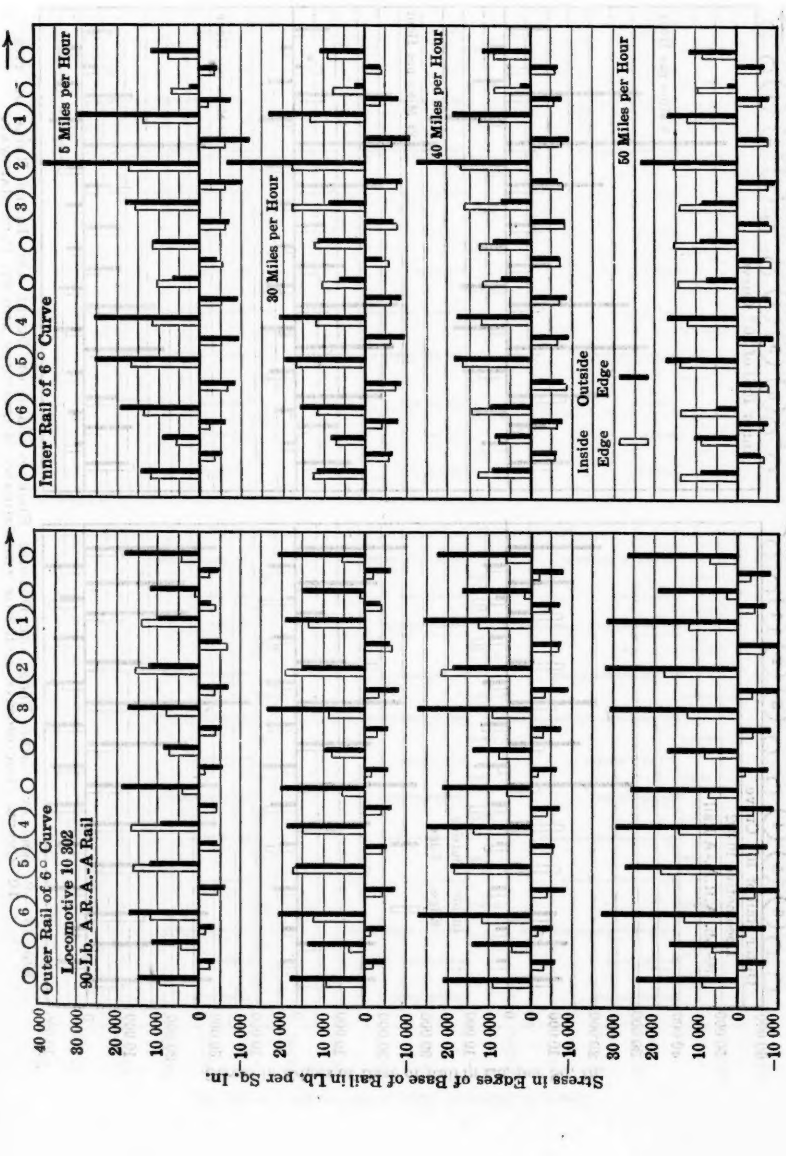


FIG. 24.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 6° CURVE. LOCOMOTIVE 10302 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

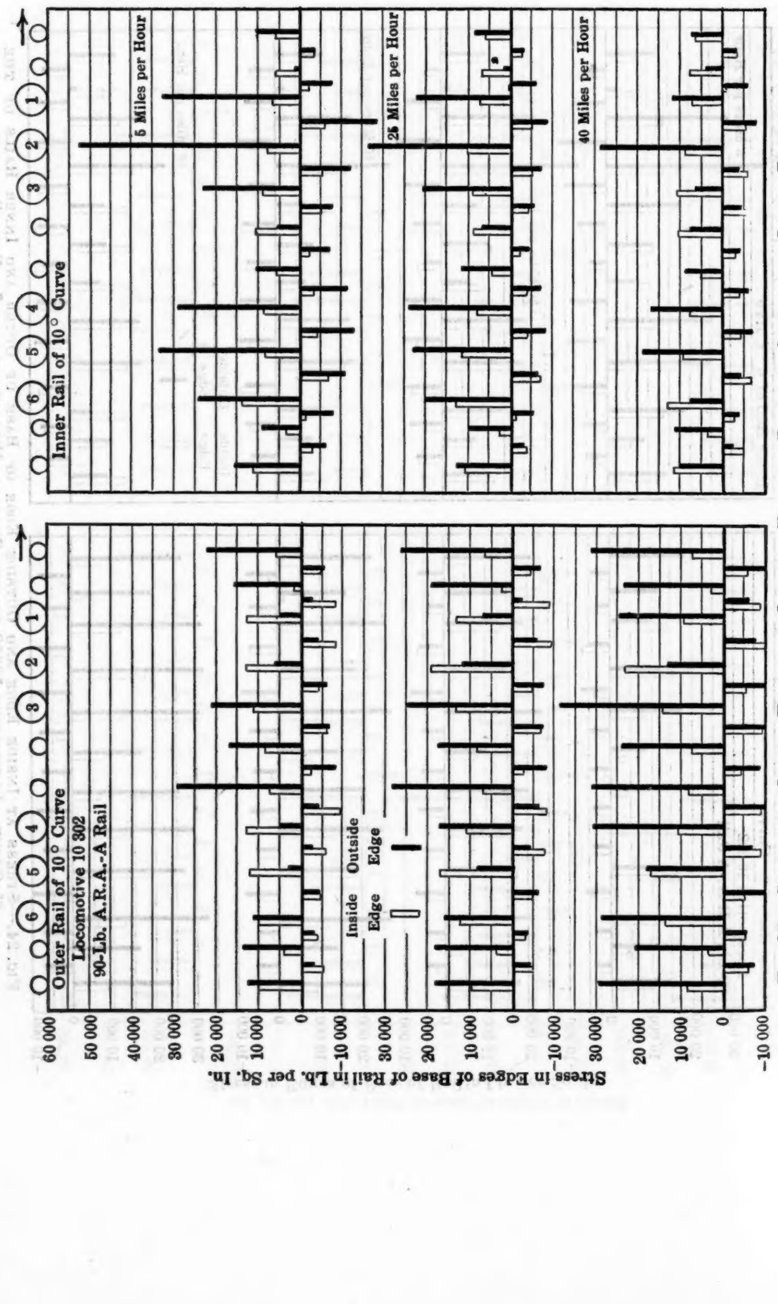


FIG. 25.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 10° CURVE. LOCOMOTIVE 10302 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

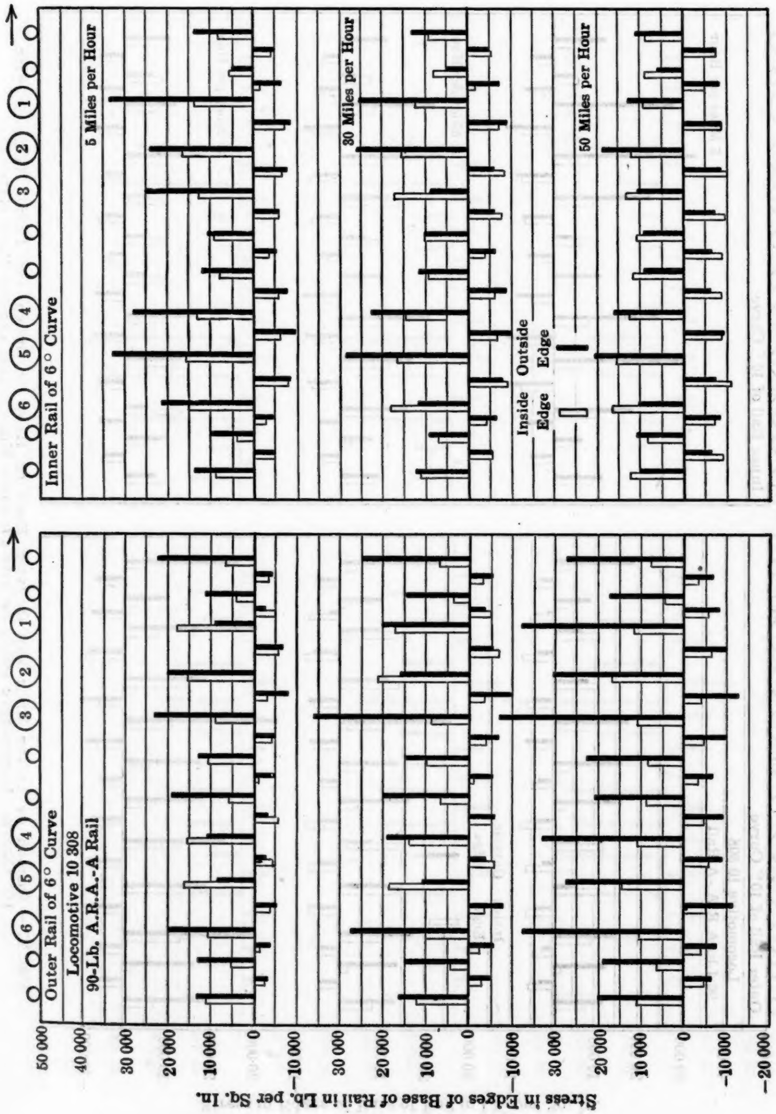


FIG. 26.—STRESSES AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 6° CURVE. LOCOMOTIVE 10308 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

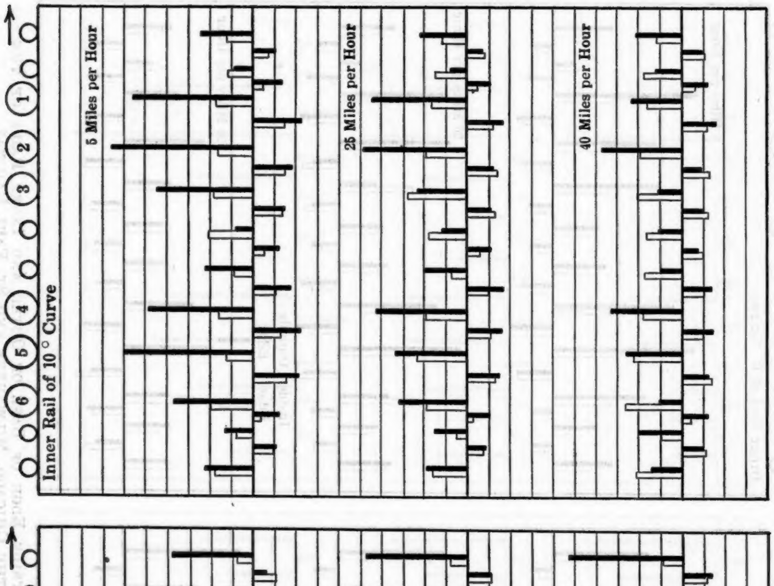


FIG. 27.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 10° CURVE. LOCOMOTIVE 10308 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

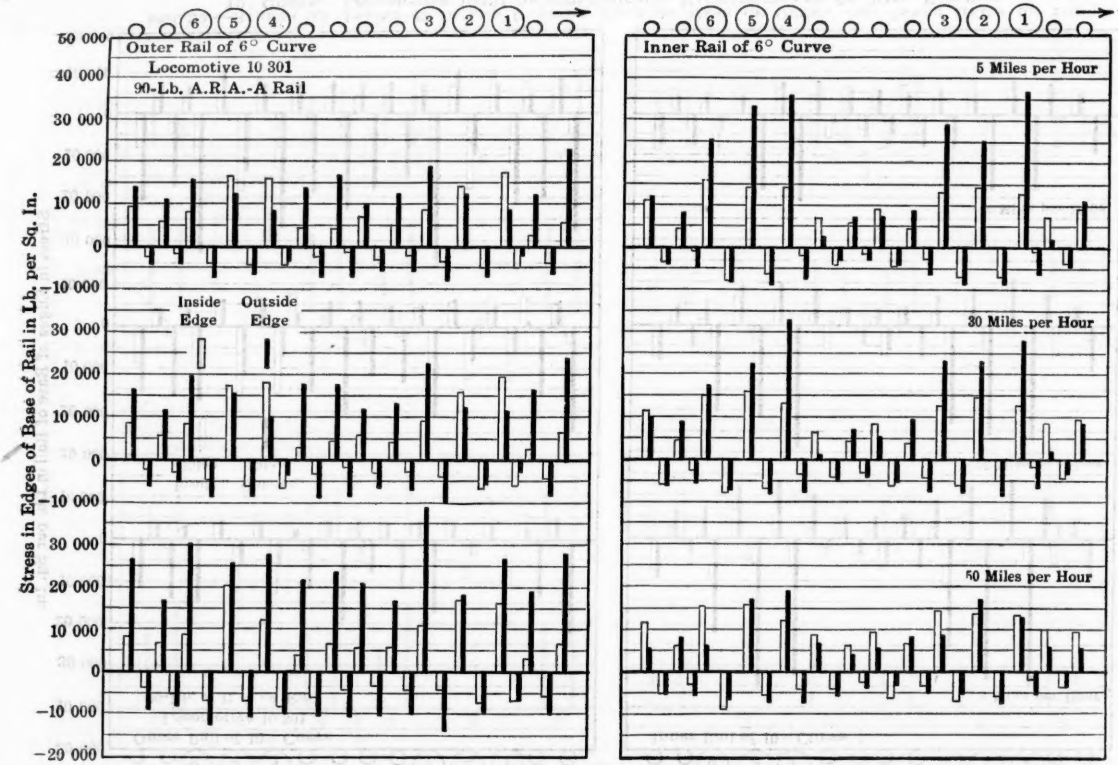


FIG. 28.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 6° CURVE. LOCOMOTIVE 10301 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

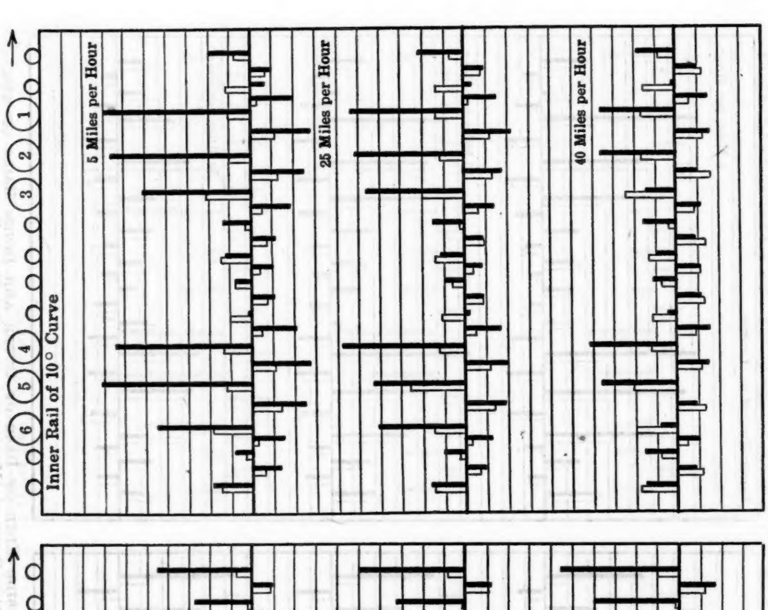


FIG. 29.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 10° CURVE. LOCOMOTIVE 10301 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

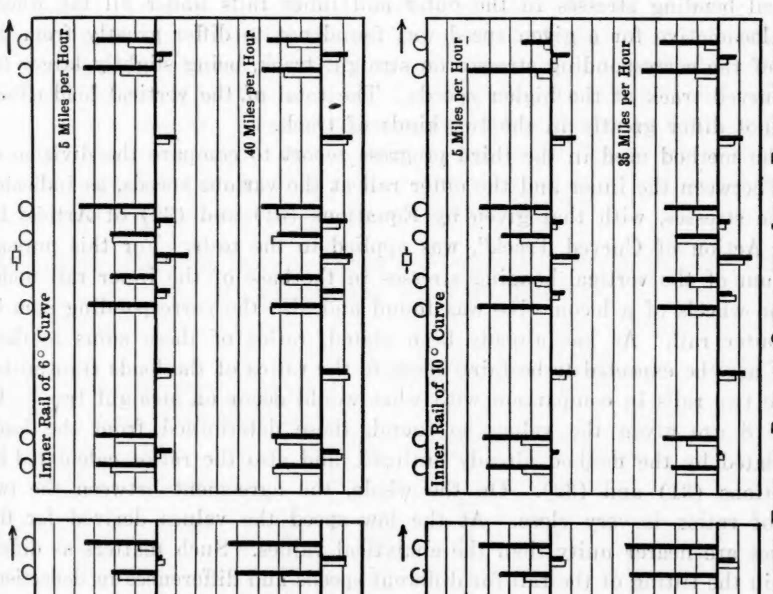


FIG. 30.—STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF OUTER AND INNER RAILS OF THE 6° AND 10° CURVES. LOADED FREIGHT CARS OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

the two rails as it is found on the various curves. A discussion will first be given based on the average stresses in the two rails, followed by one based on the loads at the individual wheels and the stresses in the rails.

As in the tests reported in the third progress report, the sum of the vertical bending stresses in the outer and inner rails under all the wheels of a locomotive for a given speed was found not to differ greatly from the sum of the corresponding stresses on straight track, being slightly larger for the curved track at the higher speeds. The total of the vertical load effects does not differ greatly in the two kinds of track.

The method used in the third progress report to compare the division of loads between the inner and the outer rail at the various speeds, as indicated by the stresses, with that given by Equations (31) and (32) of Article 10, "The Action of Curved Track", was applied to the tests. For this purpose the sum of the vertical bending stresses in the base of the inner rail under all the wheels of a locomotive was found and also the corresponding sum in the outer rail. As has already been stated, ratios of these sums to their mean may be expected to be fairly close to the ratios of the loads transmitted to the two rails in comparison with what would occur on straight track. In Table 8 are given the values so found, those determined from the loads calculated by the method already outlined, and also the ratios calculated by Equations (31) and (32). On the whole, the agreement between the two sets of ratios is very close. At the low speed the values derived for the stresses are nearer unity than the analytical values. Such matters as difference in the tilting of the rail for different speeds and differences in deflections of the springs and action of the equalizing system on the two sides of the locomotive are elements not considered in the derivation of Equations (31) and (32). It will be noted that the weights on the outer rail at the highest speeds used on the two curves are about 25% greater than that normally coming on one rail on straight track. Although these speeds, 50 miles per hour on the 6° curve and 40 miles per hour on the 10° curve, were one and two-thirds times the speed corresponding to the super-elevation of the track, the locomotives ran the curves smoothly except where the alignment was not good. The effect of the low and the high speeds on the track structure is best judged from the lateral bending stresses in the rail, which will be discussed in Article 13. It will be noted that there is little difference in the ratios obtained direct from the stresses and those obtained through the load calculated from the stresses.

The vertical bending stress in base of rail (mean of the stresses in the two edges of the base and the average of all the observations with the four instruments in all the runs) under the various wheels of the locomotive and cars is recorded in Figs. 31 to 37 for both outer and inner rails for the speeds used in the tests on curved track. For convenience, the 6° curve is shown as curving to the right instead of to the left. Comparison may be made with the stresses under the same wheels in the tests on straight track, as given in Figs. 6 to 12 and in Tables 2, 3, and 4. In making a study of the vertical bending stresses it should be kept in mind that changes in the values of

the vertical bending stresses at any wheel must be due principally to changes in the amount of load on the wheel either by itself or in combination with adjoining wheels.

The method already described was used for estimating the wheel loads necessary to produce the vertical bending stresses observed on curved track. For the sake of uniformity the results of the calculations for 5 miles per hour were changed proportionately to make the sum of the wheel loads equal the total weight of the locomotive. The factor used ordinarily differed from unity by 5% or less. For the other speeds the same factor was applied, so that the resulting load may be said to represent equivalent loads which would produce the stresses if applied statically or at a low speed as were found with the locomotive at the given higher speeds.

TABLE 8.—Ratio of Load on Inner and Outer Rail to Mean Load, Chicago, Milwaukee and St. Paul Railway.

Locomotive.	Degree of curve.	Super-elevation, in inches.	Speed, in miles per hour.	Analytical Values.		Values from Calculated Loads.		Ratios of Stresses.	
				Inner rail.	Outer rail.	Inner rail.	Outer rail.	Inner rail.	Outer rail.
10 221	6°	3.2	5	1.11	0.88	1.10	0.90	1.10	0.90
			30	0.99	1.02	0.97	1.03	0.96	1.04
			40	0.88	1.13	0.93	1.07	0.92	1.08
10 254	6°	3.2	5	1.10	0.90	1.09	0.91	1.09	0.92
			30	0.98	1.02	0.98	1.02	0.97	1.03
			40	0.89	1.12	0.89	1.11	0.87	1.13
			50	0.76	1.26	0.88	1.12	0.87	1.13
10 302	6°	3.2	5	1.13	0.87	1.13	0.87	1.13	0.87
			30	0.99	1.02	0.99	1.01	0.99	1.01
			40	0.87	1.14	0.90	1.10	0.90	1.10
			50	0.72	1.30	0.77	1.23	0.77	1.23
10 308	6°	3.2	5	1.13	0.87	1.07	0.93	1.07	0.93
			30	0.99	1.01	0.91	1.09	0.95	1.05
			50	0.72	1.30	0.77	1.23	0.77	1.23
10 301	6°	3.2	5	1.13	0.87	1.09	0.91	1.10	0.90
			30	0.99	1.02	0.95	1.05	0.97	1.03
			50	0.72	1.30	0.73	1.27	0.74	1.26
10 321	10°	3.8	5	1.13	0.87	1.09	0.91	1.09	0.91
			25	0.99	1.02	0.95	1.05	0.95	1.05
			40	0.75	1.27	0.76	1.24	0.75	1.25
10 254	10°	3.8	5	1.13	0.87	1.10	0.90	1.08	0.92
			25	0.99	1.04	0.98	1.02	0.96	1.04
			40	0.76	1.25	0.79	1.21	0.79	1.21
10 302	10°	3.8	5	1.15	0.85	1.11	0.89	1.11	0.89
			25	0.99	1.02	0.93	1.07	0.92	1.08
			40	0.72	1.30	0.70	1.20	0.70	1.30
10 308	10°	3.8	5	1.15	0.85	1.11	0.89	1.11	0.89
			25	0.99	1.02	0.96	1.04	0.95	1.05
			40	0.72	1.30	0.74	1.26	0.74	1.26
10 301	10°	3.8	5	1.15	0.85	1.07	0.93	1.09	0.91
			25	0.99	1.02	0.93	1.07	0.93	1.07
			40	0.72	1.30	0.72	1.28	0.72	1.27
8 675	10°	3.8	5	1.16	0.84	1.14	0.86	1.12	0.88
			25	0.99	1.02	0.97	1.03	0.96	1.04
			35	0.82	1.20	0.82	1.18	0.80	1.20

In Tables 9, 10, and 11 are given the calculated vertical wheel loads on straight track and on the outer and inner rails of the curves for all the locomotives with the exception of Locomotive 10301 and for the cars at several speeds, as found by the method just described. It will be noted that the distribution of load on straight track frequently differs from the nominal loads.

It is also seen that in many cases there are transfers of load from wheel to wheel, sometimes quite marked though not as strongly as in the tests of the locomotive reported in the third progress report.

The vertical bending stresses will now be considered for each locomotive and some discussion made of the results.

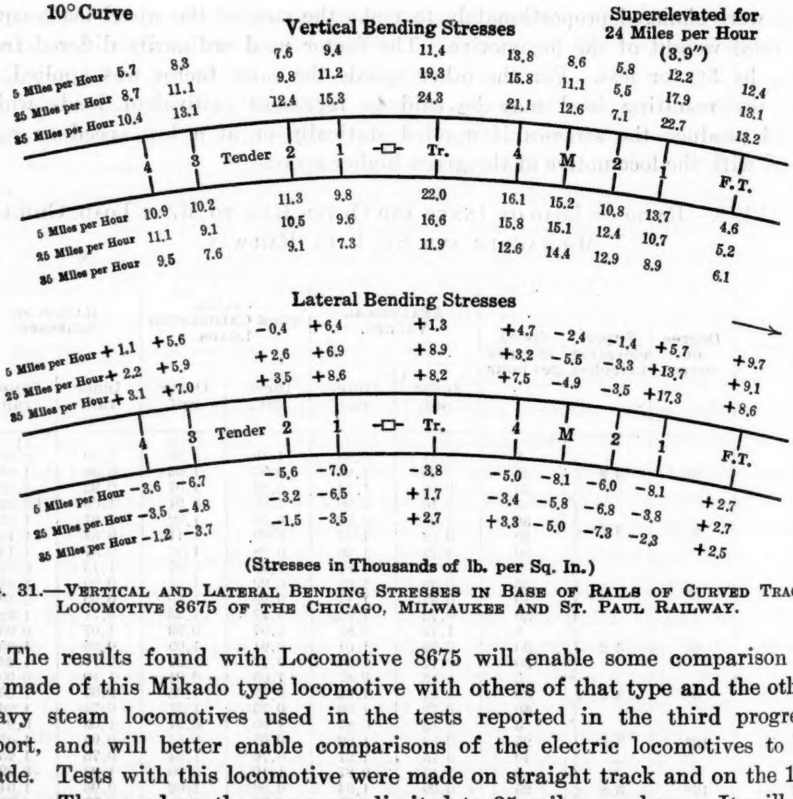
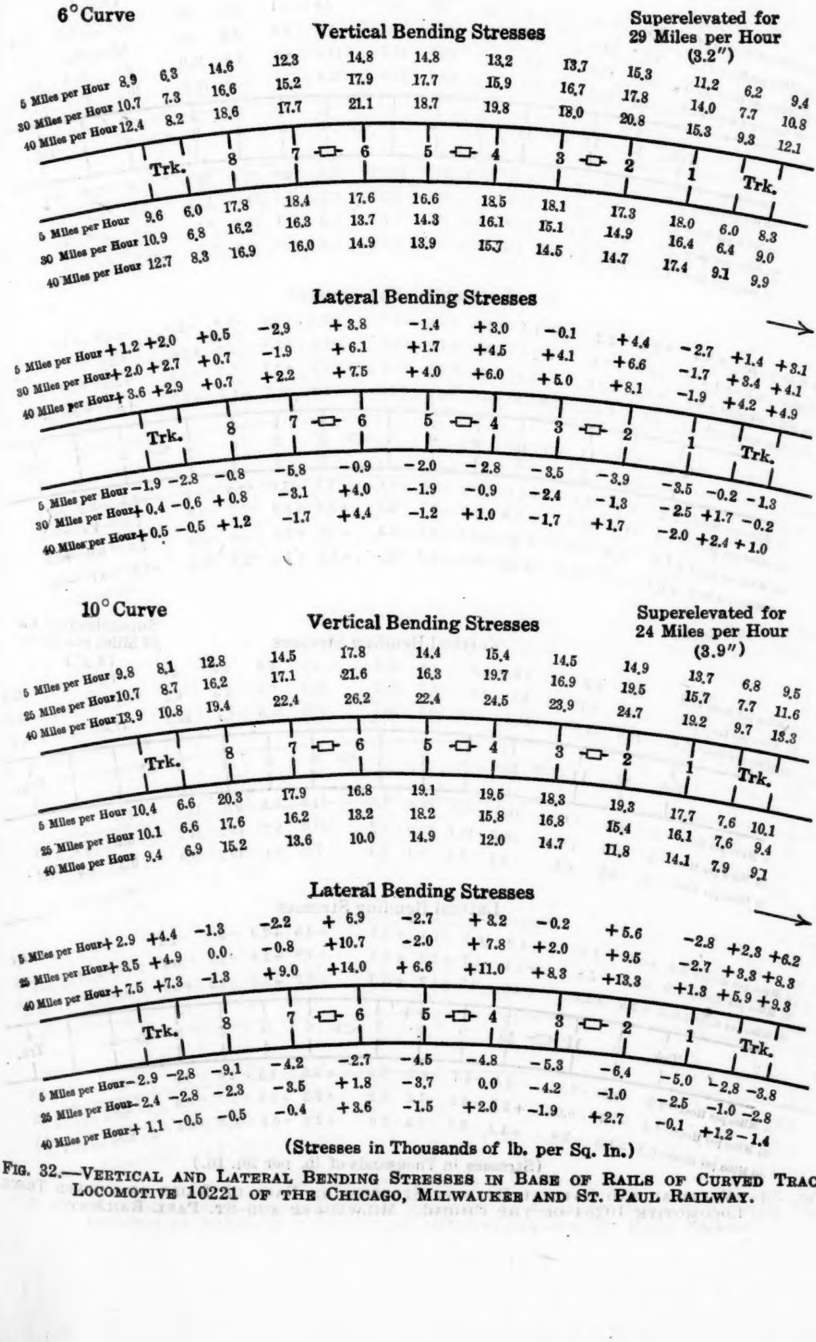


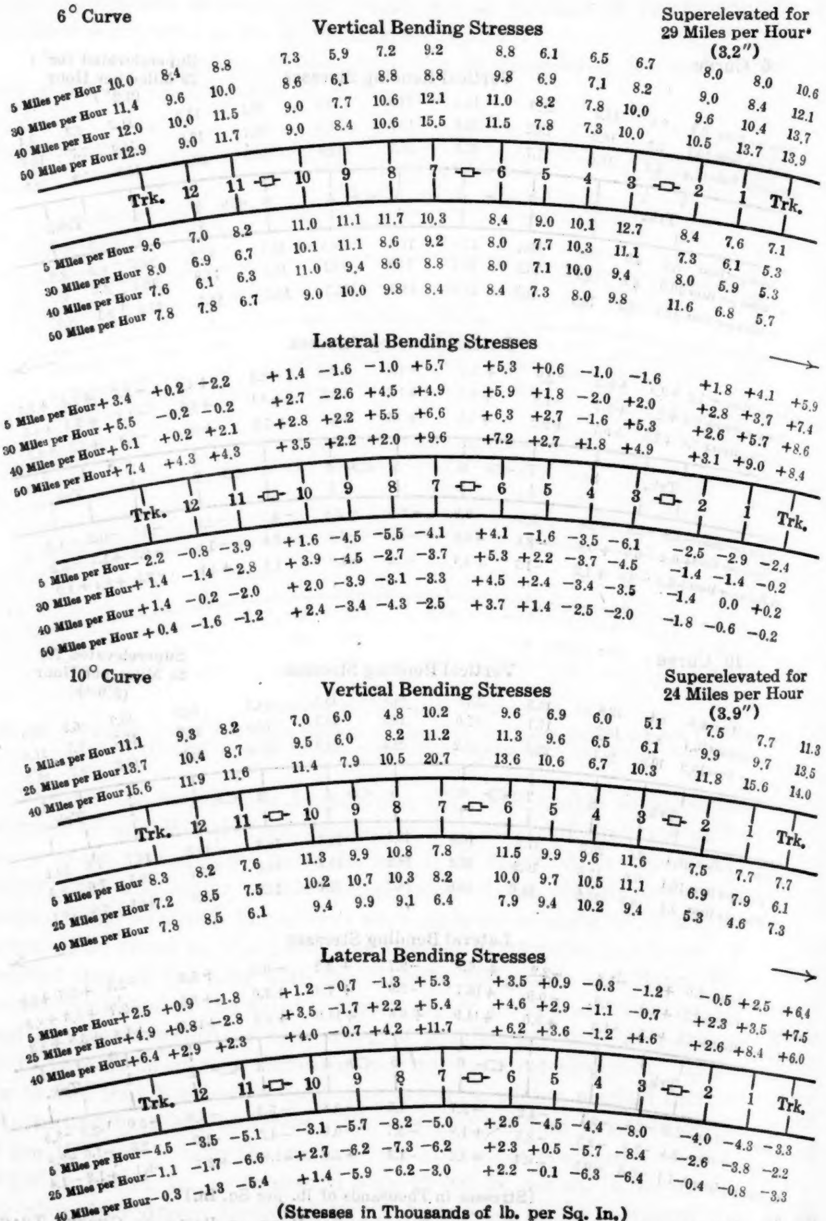
FIG. 31.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF CURVED TRACK. LOCOMOTIVE 8675 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

The results found with Locomotive 8675 will enable some comparison to be made of this Mikado type locomotive with others of that type and the other heavy steam locomotives used in the tests reported in the third progress report, and will better enable comparisons of the electric locomotives to be made. Tests with this locomotive were made on straight track and on the 10° curve. The speed on the curve was limited to 35 miles per hour. It will be recalled that all the tests with the Mikado, Mountain, and Santa Fé types of locomotive given in the third progress report showed an excessive increase of load on one inner driver at a speed of 5 miles per hour and even at speeds up to two-thirds of that corresponding to the super-elevation. It is interesting to find that Locomotive 8675 does not show such a marked increase of load transmitted by a single driver. The vertical bending stress in the inner rail of the 10° curve under the main driver at 5 miles per hour (see Fig. 31) was 15 200 lb. per sq. in., as compared with 12 400 lb. per sq. in. on straight track; the rear inner driver developed 16 100 lb. per sq. in. as compared with 14 100 lb. per sq. in. on straight track. As calculated from the stresses developed at 5 miles per hour, the weight on the main driver on straight track was 31 800 lb. (considerably higher than those on the first two drivers), and that on the inner rail of the 10° curve was 39 400 lb. It will be recalled



(Stresses in Thousands of lb. per Sq. In.)

FIG. 32.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF CURVED TRACK. LOCOMOTIVE 10221 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.



(Stresses in Thousands of lb. per Sq. In.)

FIG. 33.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF CURVED TRACK. LOCOMOTIVE 10254 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

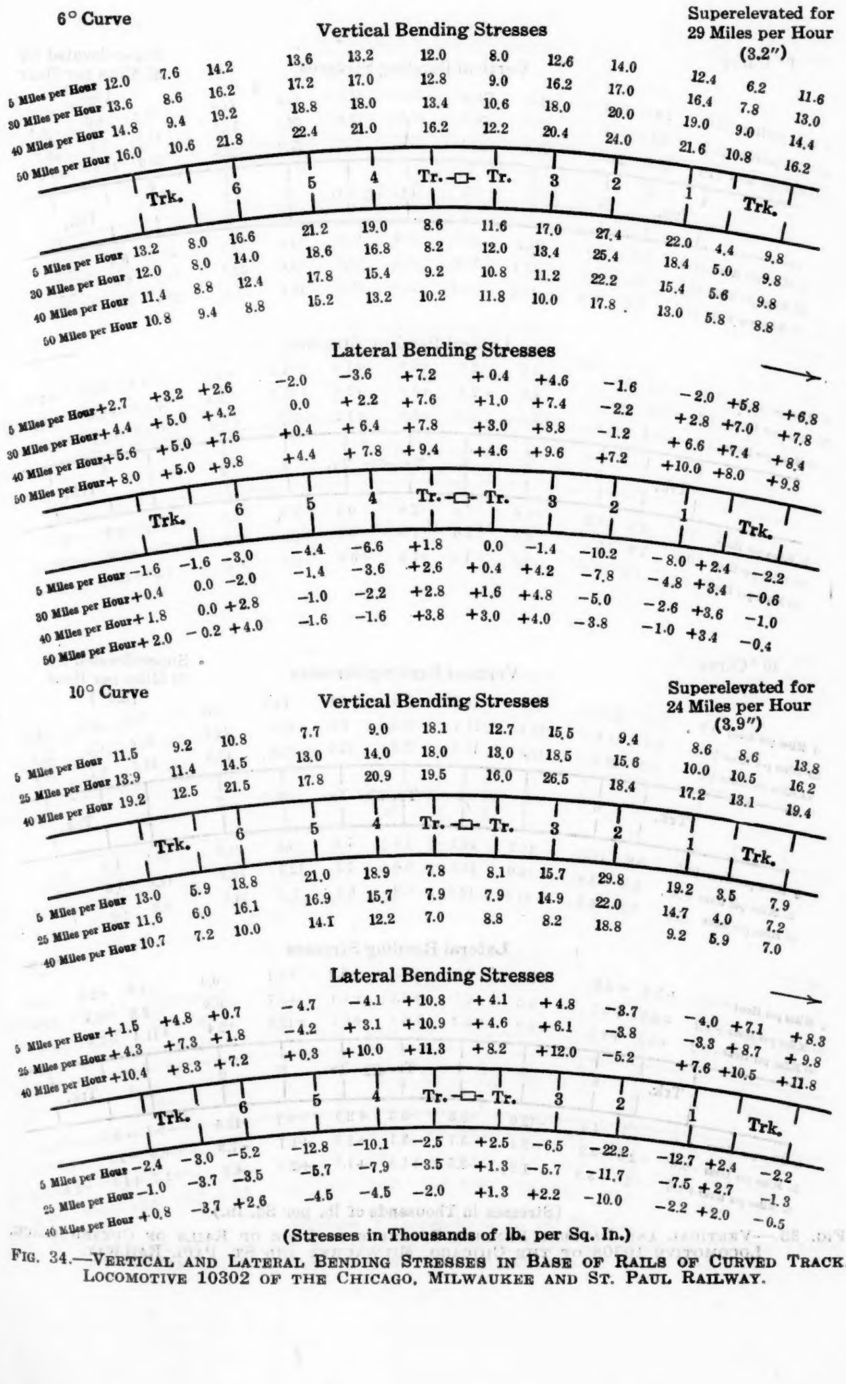
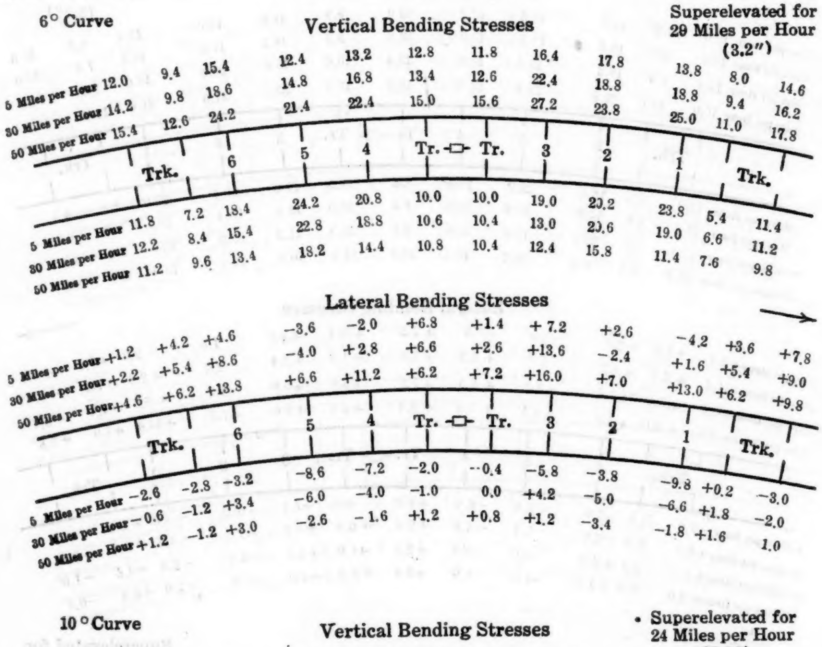


Fig. 34.—Vertical and Lateral Bending Stresses in Base of Rails of Curved Track. Locomotive 10302 of the Chicago, Milwaukee and St. Paul Railway.



(Stresses in Thousands of lb. per Sq. In.)

FIG. 35.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF CURVED TRACK. LOCOMOTIVE 10308 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

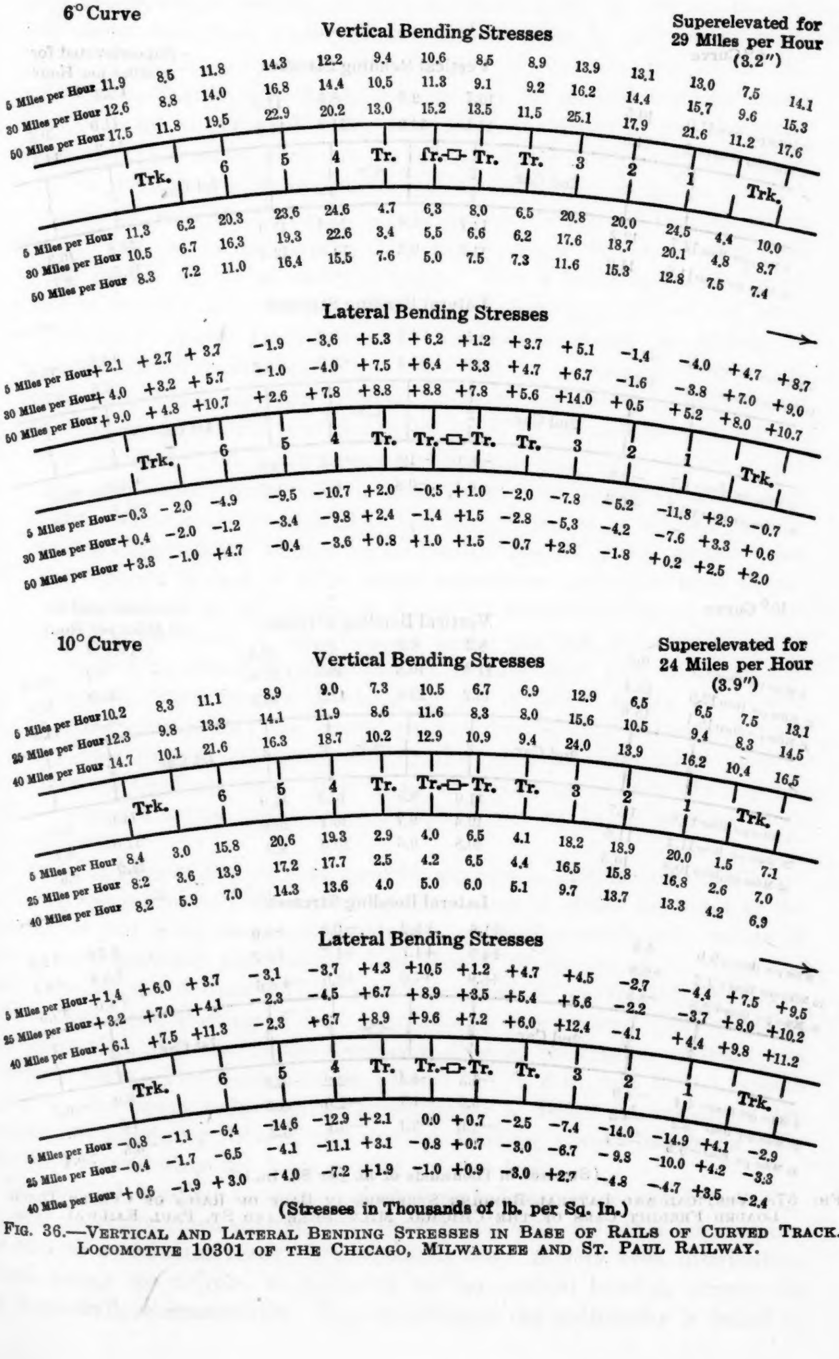


FIG. 36.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF CURVED TRACK. LOCOMOTIVE 10301 OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

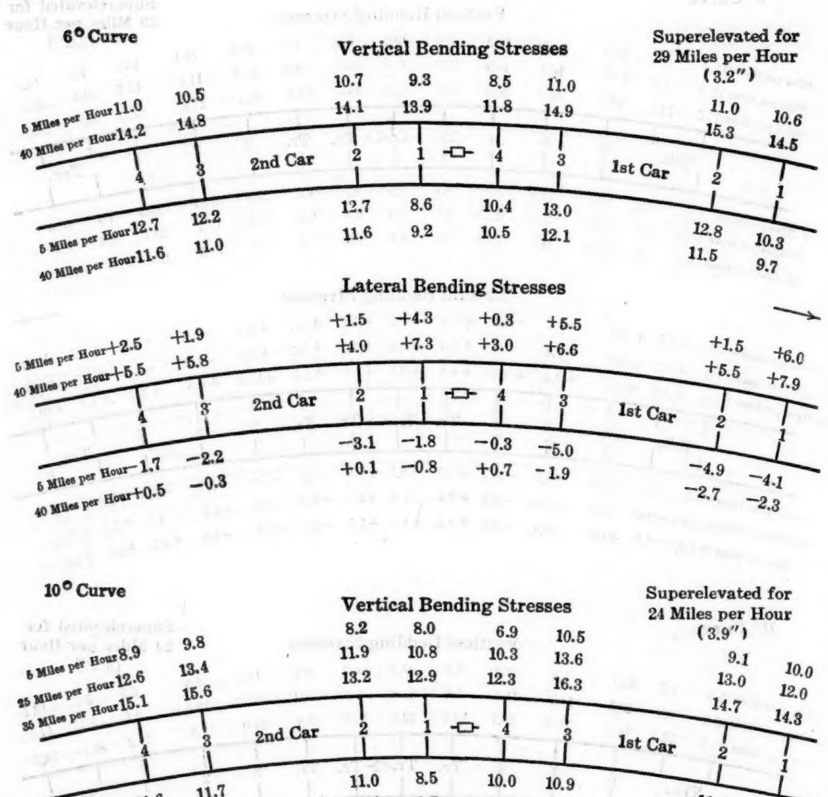


FIG. 37.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF CURVED TRACK. LOADED FREIGHT CARS OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

that on the $7\frac{1}{2}°$ curve of the Delaware, Lackawanna and Western Railroad, with 105-lb. rail, a vertical bending stress of 18 900 lb. per sq. in. was developed under the third driver of the Mikado type locomotive as compared with 11 100 lb. per sq. in. on straight track. On the 90-lb. rail of the Atchison, Topeka and Santa Fé Railway, a vertical bending stress of 26 700 lb. per sq. in. occurred under the third inner driver of the Mountain type locomotive at 5 miles per hour on the 10° curve and one of 19 700 lb. per sq. in. on the 6° curve, as compared with 13 300 lb. per sq. in. on straight track. Similarly, with the Santa Fé type locomotive a vertical bending stress of 29 200 lb. per sq. in. occurred at the fourth inner driver, as compared with 12 200 lb. per sq. in. on straight track. Why a large increase of load should not be found under one of the drivers of Locomotive 8675 on the inner rail at the lower speeds is not known. It will be shown in Article 13 that the lateral bending stresses in the inner rail for a speed of 5 miles per hour are smaller under all four drivers of this locomotive than were found with the Mikado, Mountain, and Santa Fé types of locomotive used in the earlier tests on curved track. Whether a difference lies in the design of the springs and equalizing system and in other features of the construction that would give a better division of load is not known, but it is evident that in some way the adjustability of this locomotive on the 10° curve at the lower speeds was superior to that of other steam locomotives that have been tested. For the outer rail of the 10° curve, it should be noted that the greatest vertical bending stress under a driver occurs under the first outer driver at 35 miles per hour (the lateral bending stresses showing that this driver at this speed contributes more largely to changing the direction of the locomotive than the track wheel), that the second pair of drivers carries less load than the others, and that at all speeds one or the other of the trailers gives a higher vertical bending stress than is to be found under any driver, the greatest vertical bending stress being 24 300 lb. per sq. in. at a speed of 35 miles per hour.

A characteristic of the vertical bending stresses in Locomotive 10221 (see Fig. 32) is the uniformity of vertical bending stress in either the outer or the inner rail at any speed under all the drivers of the locomotive, the values at a given speed generally not varying more than 2 000 lb. per sq. in. from the average for all the drivers. This is true for both 6° curve and 10° curve. There is then little shifting of load from axle to axle or even from wheel to wheel other than that due to the combined effect of inclination of track and centrifugal force. The spacing of the drivers (10 ft. 6 in. and 11 ft.) is such as not to give much reduction in stress under a driver by reason of the proximity of adjoining drivers, the vertical bending stress thus being relatively high considering the loads carried by the drivers.

The vertical bending stresses developed with Locomotive 10254 (see Fig. 33) were generally much smaller than those found with the other locomotives. The stresses on straight track were especially low. A very even distribution of load among the drivers, as indicated by the vertical bending stresses in rail, is generally characteristic. The exception to the uniformity is found in

TABLE 9.—CALCULATED WHEEL LOADS THAT WOULD PRODUCE STATICALLY THE MEASURED VERTICAL BENDING STRESSES FOR THE LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

Speed, in miles per hour.	Rear End Truck.		Driver.								Head End Truck.	
	2.	1.	8.	7.	6.	5.	4.	3.	2.	1.	2.	1.
					Locomotive 10221.							
Straight Track:												
Nominal loads	15 600	15 400	28 300	27 200	29 800	27 600	25 100	27 300	28 700	29 600	15 000	15 100
5	16 300	17 800	28 500	26 600	26 800	27 000	26 300	26 500	26 700	27 400	18 400	16 400
25	17 100	18 600	28 900	27 700	27 100	27 400	26 600	27 000	26 600	28 800	19 600	17 300
40	17 700	19 500	30 100	28 400	28 800	28 700	27 400	28 000	27 900	28 600	19 600	17 900
6° Curve:												
5 Inner	17 200	18 200	31 900	31 200	29 900	28 500	31 200	30 700	29 800	31 900	17 900	15 800
Outer	16 000	17 300	26 200	21 700	24 900	24 800	22 700	23 300	25 300	21 600	16 400	16 500
30 Inner	19 200	19 800	29 500	27 600	23 700	24 400	26 900	25 800	25 800	29 400	18 100	16 300
Outer	19 100	20 100	30 100	26 500	30 100	29 900	27 300	28 400	29 800	26 700	20 000	19 200
40 Inner	22 400	22 400	31 000	27 400	25 300	23 900	26 300	24 900	25 600	31 600	22 900	18 500
Outer	22 000	22 700	33 900	30 800	35 000	32 200	33 300	31 200	34 400	29 700	23 400	21 700
10° Curve:												
5 Inner	17 300	18 700	33 200	28 600	27 000	29 800	30 500	29 100	30 300	30 000	19 200	17 000
Outer	16 500	18 400	22 700	23 100	27 200	23 200	24 100	23 000	23 500	23 600	16 800	15 700
25 Inner	16 700	17 800	28 600	25 700	21 800	27 800	25 200	26 300	24 700	27 200	18 600	16 000
Outer	18 100	20 700	28 000	27 400	32 900	26 700	30 300	27 200	30 200	27 400	19 400	19 100
40 Inner	15 700	17 300	25 500	20 500	18 100	22 500	19 500	22 500	19 500	24 000	18 300	15 500
Outer	23 100	25 600	34 000	35 600	40 300	36 000	38 400	37 800	38 400	33 300	23 900	22 100

TABLE 9.—(*Continued.*)

Speed, in miles per hour.	Rear End Truck.	Driver.												Head End Truck.
	1.	12.	11.	10.	9.	8.	7.	6.	5.	4.	3.	2.	1.	1.
							Locomotive 10254.							
Straight Track:														
Nominal loads...	15 000	18 500	18 600	23 700	23 700	22 200	22 900	22 200	23 200	23 700	23 800	17 900	19 000	16 300
5........	16 600	17 700	20 900	22 800	21 700	21 900	23 700	22 600	21 700	21 200	22 800	21 300	18 500	17 600
25........	16 100	17 100	20 000	21 900	20 700	21 300	22 800	21 800	20 700	20 600	22 200	20 500	17 800	17 700
40........	17 200	18 300	21 300	22 800	21 800	22 200	23 300	22 800	21 800	21 300	23 000	22 500	18 400	17 700
60........	19 100	21 300	24 300	26 600	25 700	25 800	26 900	27 700	26 600	25 600	25 800	24 400	20 700	19 000
6° Curve:														
5 Inner........	17 500	16 900	21 500	26 600	27 100	27 900	26 700	24 200	24 200	25 600	27 800	21 600	14 700	14 500
Outer........	18 100	19 100	19 600	19 200	17 400	18 900	21 600	21 200	17 300	17 400	18 000	19 800	17 700	19 800
30 Inner........	14 800	15 600	18 500	23 800	25 300	22 900	23 800	22 300	21 500	24 000	24 700	18 600	14 000	9 800
Outer........	20 300	20 300	21 500	21 600	18 600	21 500	21 500	23 100	19 500	19 500	21 000	21 700	19 100	21 000
50 Inner........	14 900	16 600	18 400	22 400	23 700	23 700	22 600	22 000	20 300	21 200	22 900	24 000	16 000	12 900
Outer........	22 400	21 000	26 600	24 800	24 600	27 300	33 400	28 600	23 000	22 000	24 500	26 500	27 400	25 300
10° Curve:														
5 Inner........	16 300	18 200	21 000	26 600	25 400	26 400	24 200	27 900	25 900	25 400	26 800	21 500	17 400	15 800
Outer........	19 900	19 700	20 400	18 700	17 300	16 200	22 900	22 500	18 400	17 000	16 100	18 500	17 300	19 500
25 Inner........	14 800	18 100	20 300	25 200	25 900	25 700	24 300	26 000	25 300	25 900	25 900	19 400	16 900	18 100
Outer........	24 000	22 600	23 100	23 600	19 800	22 300	26 600	26 900	23 600	19 900	20 600	23 500	21 600	23 600
40 Inner........	15 200	17 700	18 000	22 900	23 400	22 600	20 200	21 900	23 200	23 800	22 400	15 800	12 000	18 200
Outer........	27 800	26 700	29 200	29 500	26 800	30 200	42 200	34 600	28 600	28 700	27 800	29 600	30 900	26 800

TABLE 10.—CALCULATED WHEEL LOADS THAT WOULD PRODUCE STATICALLY THE MEASURED VERTICAL BENDING STRESSES FOR LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

Speed, in miles per hour.	REAR END TRUCK.		DRIVER.			TRAILER.		DRIVER.			HEAD END TRUCK.	
	2.	1.	6.	5.	4.	2.	1.	3.	2.	1.	2.	1.
						LOCOMOTIVE 10302.						
Straight track:												
Nominal loads	17 000	17 000	33 000	29 900	31 400	25 300	25 000	31 800	35 300	34 000	14 500	14 500
5	20 000	18 800	28 600	31 500	32 000	25 000	21 400	30 600	37 500	28 600	17 500	17 500
25	20 300	19 700	29 300	31 800	32 200	25 300	22 100	31 000	38 600	29 300	17 400	18 600
40	21 700	20 800	30 800	33 100	33 200	26 400	23 400	32 200	39 600	31 300	19 800	20 400
60	26 600	26 800	35 800	37 500	37 800	31 800	28 500	36 400	42 600	39 900	23 300	22 200
6° Curve:												
5 Inner	22 300	21 000	81 500	37 700	35 100	22 100	25 400	34 900	46 600	38 600	16 800	17 300
Outer	20 100	19 000	26 300	25 700	25 800	23 600	19 100	24 200	25 500	23 400	16 400	19 000
30 Inner	20 800	19 700	27 300	33 200	31 300	20 800	24 600	29 800	42 300	33 400	16 500	17 100
Outer	22 900	21 600	30 400	32 000	32 400	26 400	22 000	30 400	31 300	30 300	20 100	21 800
50 Inner	18 500	19 600	19 700	26 800	25 800	22 100	23 300	22 600	30 300	24 600	14 600	15 300
Outer	27 400	27 000	40 100	41 500	40 600	33 600	28 800	39 300	43 400	40 100	27 300	27 700
10° Curve;												
5 Inner	22 400	19 300	35 100	39 200	36 000	20 900	20 900	33 800	50 500	36 100	14 800	14 800
Outer	20 100	20 500	21 600	17 000	21 200	33 000	28 300	29 500	20 200	19 100	19 800	22 900
25 Inner	20 200	18 000	30 200	82 000	30 200	19 800	19 800	30 400	38 400	28 000	13 500	13 300
Outer	24 700	25 900	29 000	26 600	29 900	34 900	30 200	35 600	30 300	23 400	23 700	27 100
40 Inner	18 400	17 400	21 100	26 000	24 100	17 100	18 500	20 000	31 200	20 000	14 400	12 900
Outer	33 300	31 400	41 200	36 800	41 900	40 500	37 700	49 000	38 100	35 600	31 100	83 300

TABLE 10.—(*Continued*).

Speed, in miles per hour.	Rear End Truck. 2.	Rear End Truck. 1.	Driver. 6.	Driver. 5.	Driver. 4.	Trailer. 2.	Trailer. 1.	Driver. 3.	Driver. 2.	Driver. 1.	Head End Truck. 2.	Head End Truck. 1.
					Locomotive 10308.							
Straight Track:												
Nominal loads	16 600	16 600	33 400	33 200	32 700	23 300	22 700	30 500	37 500	37 300	15 100	15 100
5	18 300	17 000	29 400	31 800	32 500	24 600	22 800	33 000	34 600	30 800	19 300	19 900
40	20 000	20 200	31 700	34 300	33 900	26 000	23 600	33 500	35 000	31 000	20 200	21 300
60	20 800	22 500	33 800	37 100	36 000	27 300	25 700	36 600	37 400	33 000	23 100	22 700
6° Curve:												
5 Inner	19 300	19 000	32 400	40 300	36 700	23 100	22 600	33 600	35 700	38 000	17 800	18 800
Outer	19 100	20 600	26 700	23 100	24 700	24 600	24 300	29 800	30 400	25 600	19 300	22 500
30 Inner	19 800	19 800	28 100	37 500	33 800	23 000	21 600	25 800	34 000	32 000	18 100	18 400
Outer	25 100	32 900	35 400	37 600	32 200	27 200	26 500	38 300	34 200	33 100	22 900	25 400
50 Inner	18 600	20 300	25 200	30 300	26 900	21 900	20 900	23 400	26 200	21 400	16 600	15 900
Outer	25 900	28 600	40 500	38 800	40 000	31 700	33 400	46 800	43 100	42 800	27 500	28 600
10° Curve:												
5 Inner	20 600	18 300	32 100	39 400	36 600	22 900	22 300	37 000	43 500	38 900	19 200	19 000
Outer	20 600	21 000	26 800	20 400	23 200	24 000	26 700	31 000	23 400	21 400	17 400	21 700
25 Inner	18 400	17 500	27 800	31 600	33 300	20 900	22 100	30 100	36 100	32 400	18 700	17 400
Outer	22 800	23 700	32 200	30 400	28 200	26 600	27 400	39 200	30 600	27 200	20 500	26 600
40 Inner	18 400	19 800	22 300	26 800	28 000	19 300	18 000	20 800	29 300	24 000	19 600	17 200
Outer	28 500	29 500	47 200	43 400	42 400	33 300	34 200	50 400	41 300	42 000	24 900	30 500

Locomotive 10308 shows greater variations in the vertical bending stress under the several drivers, both on straight track and on the 6° and 10° curves, than the two electric locomotives just considered. On straight track

the outer rail under Driver No. 7 (the leading driver of the second half of the locomotive) for the highest speeds on both the 6° and 10° curves, where the stress is considerably greater than under the other drivers on either rail, 15 500 lb. per sq. in. on the 6° curve, and 20 700 lb. per sq. in. on the 10° curve. It will be shown that the lateral bending stress in the outer rail under this driver (11 700 lb. per sq. in. on the 10° curve at 40 miles per hour) is much greater than that under the outer drivers, and that the flange of this driver is the principal factor in the turning action of the second unit of drivers in traversing the curve. It seems possible that the springs and equalizing system are in some way affected in their action by this strong lateral pressure.

TABLE 11.—CALCULATED WHEEL LOADS THAT WOULD PRODUCE STATICALLY THE MEASURED VERTICAL BENDING STRESSES FOR LOCOMOTIVE 8675 AND LOADED FREIGHT CARS OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

Speed, in miles per hour.	Trailer.	Driver.				Head End Truck.
		4.	Main.	2.	1.	1.
Straight Track:						
Nominal loads	28 500	30 500	30 500	29 300	29 200	12 000
5	25 200	31 300	31 800	27 600	28 200	16 000
25	27 600	33 800	33 800	29 200	29 600	17 300
40	27 200	35 200	35 300	30 800	31 200	18 400
10° Curve:						
5 Inner	35 600	36 200	39 400	31 200	29 600	10 500
5 Outer	19 400	27 900	23 600	20 000	25 800	21 400
25 Inner	27 600	35 000	36 800	32 400	25 600	10 900
25 Outer	31 400	33 300	29 600	23 100	36 500	23 400
35 Inner	20 200	29 100	34 400	31 900	23 000	11 800
35 Outer	39 600	43 400	35 200	27 600	43 900	24 800

	Second Car.				First Car.			
	4.	3.	2.	1.	4.	3.	2.	1.
Straight Track:								
Nominal loads	19 000	19 000	18 800	18 800	19 700	19 700	19 000	19 000
5	18 800	18 500	19 100	19 600	19 600	19 500	19 000	18 700
40	20 100	20 100	21 700	23 700	23 700	22 000	22 100	22 000
6° Curve:								
5 Inner	19 900	19 400	20 200	18 800	20 900	21 000	20 500	20 000
5 Outer	17 300	16 700	17 600	18 600	17 800	17 900	18 400	20 700
40 Inner	18 100	17 600	18 900	19 400	22 800	19 800	18 800	19 600
40 Outer	22 800	23 200	23 800	27 000	24 800	24 300	25 400	28 000
10° Curve:								
5 Inner	15 800	19 500	22 400	22 000	20 000	19 900	22 000	25 200
5 Outer	16 000	17 000	15 400	17 600	16 700	18 600	17 200	20 400
25 Inner	20 200	20 300	19 500	21 800	23 200	20 000	20 000	20 000
25 Outer	22 500	23 400	22 000	24 200	24 000	24 500	23 900	25 600
35 Inner	18 900	18 300	19 900	21 500	21 700	17 900	21 200	20 100
35 Outer	26 700	27 400	24 900	28 600	28 500	29 300	27 400	30 300

Locomotive 10302 shows greater diversity in the vertical bending stress under the several drivers, both on straight track and on the 6° and 10° curves, than the two electric locomotives just considered. On straight track

Driver No. 2 on both rails carries a considerably greater load than the adjacent drivers, 30% more as calculated from the stresses. The vertical bending stress in the inner rail under this driver for a speed of 5 miles per hour is 27 400 lb. per sq. in. on the 6° curve and 29 800 lb. per sq. in. on the 10° curve (the latter being indicative of a wheel load of 49 400 lb.), both vertical bending stresses being relatively very great, and that in the outer rail on the 6° curve under the corresponding driver for a speed of 50 miles per hour is 24 000 lb. per sq. in., while on the 10° curve the highest vertical bending stress in the outer rail, at 40 miles per hour (26 500 lb. per sq. in.), is shifted to Driver No. 3. The middle drivers of the second half of the locomotive do not develop as great stresses as the corresponding drivers of the first half.

For Locomotive 10308 the loads are more nearly equally divided among the drivers, as shown by the vertical bending stresses on straight track, than is the case with Locomotive 10302. At 5 miles per hour the high vertical bending stresses are in the inner rail under the middle driver of each unit of drivers (Nos. 2 and 5) for the 10° curve, the greatest being 20 500 lb. per sq. in., and under Nos. 1 and 5 for the 6° curve, 24 000 lb. per sq. in. At 50 miles per hour on the 6° curve the highest stresses are in the outer rail, 27 200, 25 000, and 24 200 lb. per sq. in., and at 40 miles per hour on the 10° curve, 22 000 and 23 500 lb. per sq. in., the highest stress being under the rear driver of a group. It is seen that the increase of wheel load on curved track, as indicated by the vertical bending stresses, except for Driver No. 2, does not differ greatly from that given by the locomotive in the original form.

For Locomotive 10301 the vertical bending stresses under the drivers on straight track indicate a good division of the load among the drivers. The greatest vertical bending stresses on curved track occur under different wheels; on the inner rail of the 10° curve under Drivers Nos. 1 and 5, at 5 miles per hour, the stresses are 20 000 and 20 600 lb. per sq. in., and on the outer rail under Drivers Nos. 3 and 6, at 40 miles per hour, 24 000 and 21 600 lb. per sq. in.; on the inner rail of the 6° curve under Drivers Nos. 1 and 4, at 5 miles per hour, 24 500 and 24 600 lb. per sq. in., and on the outer rail under Drivers Nos. 3 and 5, at 50 miles per hour, 25 100 and 22 900 lb. per sq. in. Except for the last named, the high vertical bending stresses are accompanied by the greatest of the lateral bending stresses in the rail found at the given speed. It is seen that the increase of wheel load in curved track, as indicated by the vertical bending stresses, is somewhat less than for the locomotive in the original form.

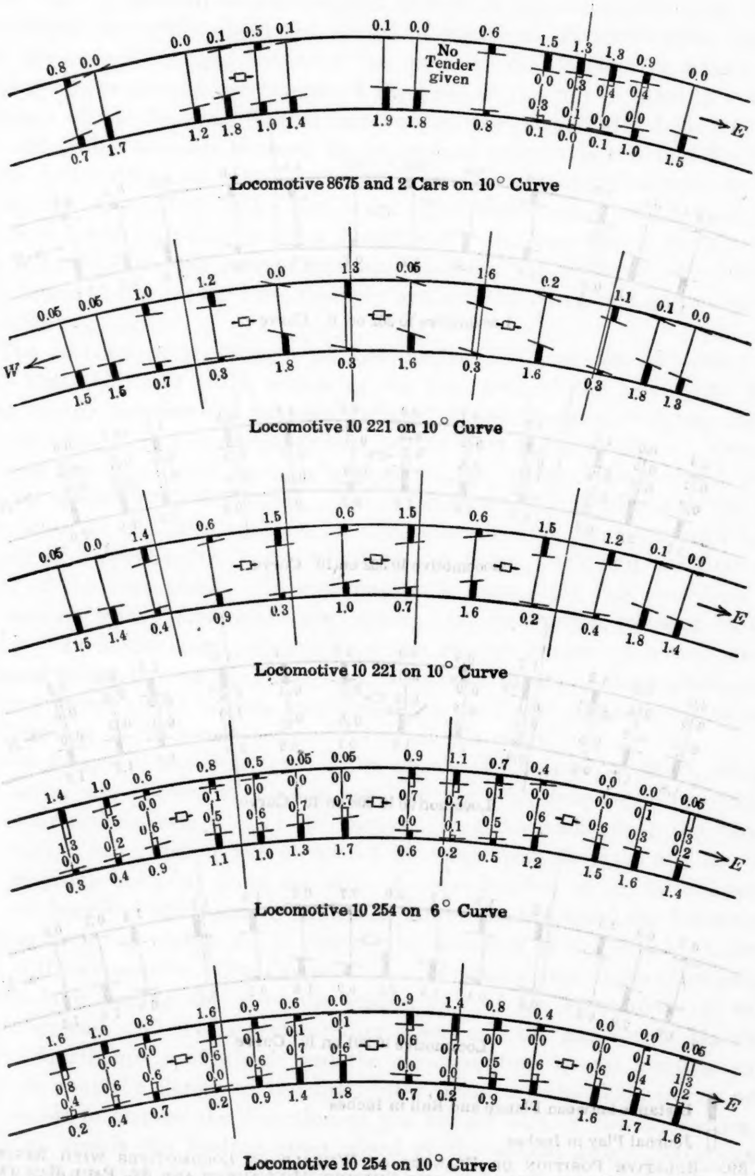
For the loaded freight cars on straight track it was found, as would be expected, that the vertical bending stresses under wheels that are close to wheels of the tender or to those of another car, are smaller than those under wheels that do not have another wheel within a few feet of it on both sides. The same influence of adjoining wheels is found on the two curves regardless of speed (Fig. 37). The vertical bending stresses on the curves are moderate, the greatest being 16 300 lb. per sq. in. on the 10° curve, at 35 miles per hour. It is evident from what has been said that with the four-

wheel truck there is little shifting of load from axle to axle on the curve, or even from wheel to wheel, except what is due to centrifugal force and inclination of track.

13.—Lateral Bending Stresses in Rail on Curved Track and Position of Flanges.—On straight track irregularities in the direction of the movement of wheels and sidewise oscillations produce lateral bending of the rail, as has already been discussed. When a locomotive or a car traverses curved track, as may be expected from analytical considerations, additional forces are applied to the rail in a lateral direction, which set up still greater lateral bending of the rail. The flange of a truck wheel or a leading driver presses against the outer rail, developing a force which changes the direction of movement of the group of wheels connected to the same frame, in the manner already described under Article 10, "The Action of Curved Track". The lateral pressures on the rail given by other wheels of the unit are most generally transmitted to the rail as either pull or push by the friction of the tread on the rail, and may be directed inwardly or outwardly of the curve according to the position in the unit and the relation of the speed to super-elevation and centrifugal force and also according to the number of wheels in the unit and even to the degree of stiffness and flexibility of the connection between units.

It may be helpful in studying the lateral bending stresses to show the position of the wheels with respect to the rail that was observed on a curve after the locomotive had come to rest without brakes being applied. In Figs. 38 and 39 is given the position of the wheels of six locomotives relative to the rail; the dimensions are given in inches and represent the distance from the gauge side of the rail to the flange of the wheel. The measurements were made at points a little above the mid-height of the head of the rail. The radial line on the diagrams is drawn at the point where the line taken by the wheel-base appears to be perpendicular to the radius of the curves. The point where this radial line crosses the inner rail may be taken to be the probable center of rotation of the locomotive frame. Here, again, the 6° curve is shown as curving to the right instead of to the left.

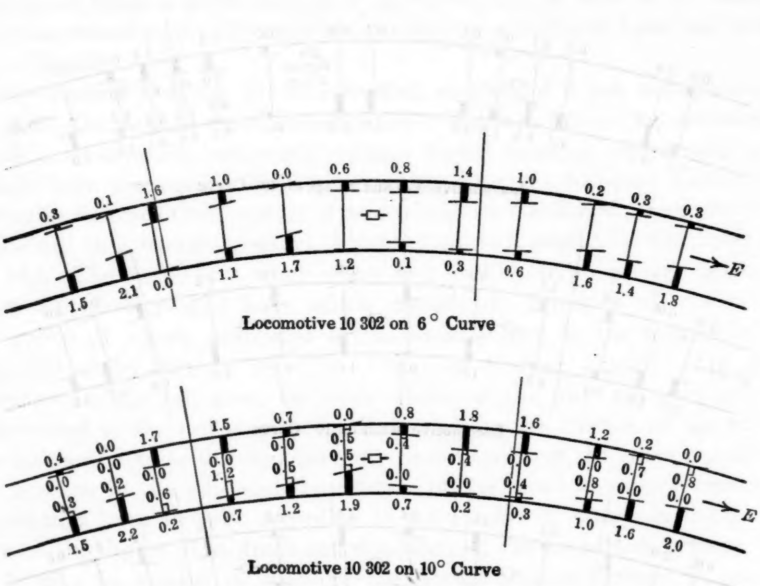
The wheels of Locomotive 8675 (Mikado type) on the 10° curve (see Fig. 38) took positions that are relatively the same as the positions found with the Mikado type locomotive on the 6° and the 7½° curves of the Delaware, Lackawanna and Western Railroad and the Santa Fé type locomotive on the 6° and the 10° curves of the Atchison, Topeka and Santa Fé Railway, reported in the third progress report. The flange of the outer front truck wheel bore against the outer rail, and that of the first outer driver was 0.9 in. away, and yet outward lateral bending of the outer rail was found under this driver at 5 miles per hour. The center of rotation appeared to be under the third inner driver. The flanges of the second, third, and fourth inner drivers were close to the inner rail and the flanges of the corresponding outer drivers were well away from the outer rail and at about the same distance from it. Measurements made on the position of the journals in the boxes showed that the journal of the first outer driver still had all its outward play available so that it was not helping in changing the direction of the locomotive. The



■ Distance between Flange and Rail in Inches

▯ Journal Play in Inches

FIG. 38.—RELATIVE POSITION OF FLANGES OF WHEELS OF LOCOMOTIVES WITH RESPECT TO THE RAILS OF CURVED TRACK ON THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.



▮ Distance between Flange and Rail in Inches

▯ Journal Play in Inches

FIG. 39.—RELATIVE POSITION OF FLANGES OF WHEELS OF LOCOMOTIVES WITH RESPECT TO THE RAILS OF CURVED TRACK ON THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

journal of the rear outer driver was at the limiting position in its bearing and, hence, was in position to put outward thrust on the outer rail as the frame was swung outwardly about the center of rotation. Measurements made to find the change in alignment of the rails as the locomotive passed at 2 miles per hour showed movements of the head of the rail in keeping with the positions of the flanges. The impressions on the copper wire laid across the rail and pulled laterally between the passage of wheels showed that the flange of the first outer truck wheel and first driver bore strongly against the outer rail at the speed of 35 miles per hour. This condition is also borne out by the high lateral bending stresses found under the two wheels at this speed. The impression on the copper wire also showed that the first outer wheel of each truck of the tender bore strongly against the outer rail at the speed of 35 miles per hour.

The position of the flanges of Locomotive 10221 on the 10° curve indicates that the outer truck wheels of the first half of the locomotive at the lower speeds exercise the turning action for the frame containing the first four wheels. The first outer drivers of the next three groups of wheels lie close to the outer rail, seemingly close to the position that would permit turning forces to be developed. The impressions of the copper wires, however, show at 25 miles per hour flange contract with the outer rail by the flanges of only the first outer wheel of the front truck and the first outer wheel of the rear truck. No explanation is known for this condition. All the runs on which impressions were taken, nine in number, agree in this particular. It would seem probable that the several groups of wheels have so adjusted themselves on the curve that, with the stiffness of the connection, the flange pressure of the leading truck wheels is the principal agency in changing the direction of the locomotive in traversing the curve.

The flanges of the first three outer wheels of Locomotive 10254, when the locomotive was allowed to come to rest (see Fig. 38), were found to be against the outer rail on both 6° and 10° curves. In the second group the flange of the leading outer driver was about 0.4 in. away from the rail, and the others of this group lay away from the rail. A consideration of the lateral bending stresses described subsequently indicates that at the low speed the flanges of the leading group of wheels act to change the direction of the wheels of the first half of the locomotive. The flange of the first outer driver of the third group of wheels (the leading driver of the second half of the locomotive) is against the outer rail, and the lateral bending stress and the position of the flanges of the fourth group of wheels and the corresponding lateral bending stresses indicate that the flange of the first outer driver of the third group acts at low speeds to change the direction of the second half of the locomotive. At the high speeds the leading outer wheel of the second group participates in the turning action, as indicated by the lateral bending stresses.

In the three forms of the Westinghouse-Baldwin locomotive, the position of the flanges of the wheels with respect to the rail, when the locomotive was allowed to come to rest, did not differ materially. The flange of the first outer truck wheel was against the outer rail of the 10° curve and that of the second outer truck wheel was close to it. The flanges of all the outer drivers were

away from the outer rail 1 in., or more, the first driver of each group being closer than the last. The wheel following the front group of drivers is also away from the outer rail, except for Locomotive 10301. The wheel in front of the rear group of drivers (and the two wheels of Locomotive 10301) is always against the outer rail, and evidently is the principal agent in changing the direction of the second half of the locomotive. The flange of the first outer wheel of the rear truck is always against the outer rail, and that of the second is close to it. The flanges of the inner drivers are usually close to the inner rail, though not against it. The impressions made with the copper wire on the outer rail at the speed of 25 miles per hour on the 10° curve indicated that the same flanges, and no others, were against the outer rail as when the locomotive came to rest. At 40 miles per hour the flange of the first outer driver of each group bore against the outer rail. On the inner rail the impressions taken at 5 miles per hour indicate that the flanges of the second and third inner drivers of each group were usually close to the rail. On the 6° curve the impressions taken with Locomotive 10308 indicate that at speeds of 30 and 50 miles per hour the flange of the first outer driver of each group of drivers bore against the outer rail; this was the only locomotive of this type for which impressions were taken on the 6° curve. In view of all the observations, it is evident that the wheels of the three forms of the Westinghouse-Baldwin locomotive took much the same position on the curves and that the modifications of the original form cannot have done more than modify the intensity of pressures. Such changes may be indicated by the lateral bending stresses in the rail. The high values of the lateral bending stress in the outer rail found under the rear driver of both halves of the locomotives at the highest speed is an indication that at high speeds the flange of these drivers may bear against the outer rail and thus that flange wear may occur on these drivers, as has been reported by the railroad company.

The impressions with copper wire made under the wheels of the loaded freight cars on the 6° curve at a speed of 40 miles per hour indicated that the flange of the first outer wheel of each truck bore against the outer rail, as also did that of the second outer wheel of the leading truck of the first car and the second outer wheel of the rear truck of the second car. These positions agree in general with the position of the wheels measured after the cars had come to rest.

In considering the lateral bending of the rails of curved track, it should be kept in mind that the beam strength of the rail in a lateral direction is much less than that which resists bending in a vertical direction (the section modulus about a vertical axis being only about one-fifth of that about a horizontal axis), large lateral bending stresses will be developed by a lateral bending moment relatively much smaller than the vertical bending moment that causes vertical bending stresses of the same magnitude. The value of the lateral bending stress that is reported is the amount by which the stress in one edge of the base of rail exceeds the mean of the stresses in the two edges (the latter being the stress called the vertical bending stress). For both outer and inner rail the stress is called positive if it tends to increase the curvature in the rail and negative if it tends to straighten the rail. The maximum fiber stress

in the base of rail will be equal, of course, to the numerical sum of the vertical bending stress and the lateral bending stress.

It will be recalled that in locomotives of the Mikado type it was reported in the third progress report that the lateral force necessary to change the direction of the locomotive in traversing a curve was developed by the outer rail against the flange of the outer truck wheel and generally also of the first driver, and that the pressure against the flange of the first driver increased considerably with an increase in speed. The outward lateral bending stresses in the rail under these two wheels thus developed by these lateral turning forces are among the greatest lateral bending stresses found. With Locomotive 8675 the lateral bending stress in base of rail under the first outer driver (see Fig. 31) ranged from 5 700 lb. per sq. in. at 5 miles per hour, to 17 300 lb. per sq. in. at 35 miles per hour. The latter stress is higher than was obtained on the outer rail of the 10° curve at a speed of 40 miles per hour with any of the electric locomotives. The lateral bending stress in the outer rail under the truck wheel remains nearly constant at the three speeds, which indicates that the restrictive action of the connection of the leading truck to the frame of the drivers is limited and that at the highest speed the first outer driver participates strongly in the guiding action. At the highest speed the flange of the first outer driver cut off the copper wire at almost every attempt to obtain a record. As is usual with this type of locomotive, the rear driver and the trailer also produced outward bending in the outer rail at all speeds. On the inner rail the higher lateral bending stresses at a speed of 5 miles per hour were under the first and third drivers, 8 100 lb. per sq. in. at each point. These stresses are relatively considerably smaller than those found with Mikado, Mountain, and Santa Fé types of locomotive reported in the third progress report. They are about the same as were developed with the General Electric freight and passenger locomotives and less than those found with Westinghouse-Baldwin locomotives. It has already been stated that this steam locomotive did not develop an excessive increase of load on one of the inner drivers at the lower speeds, as was generally the case with the locomotives having eight and ten drivers reported in the third progress report.

With Locomotive 10221 the lateral bending stresses in the inner rail are relatively small (see Fig. 32), the greatest being 9 100 lb. per sq. in. under one driver on the 10° curve and 5 800 lb. per sq. in. on the 6° curve at 5 miles per hour, all the other values being considerably smaller. In the outer rail the stresses increase with an increase in speed. At 40 miles per hour the greatest lateral bending stress in the outer rail on the 6° curve is 8 100 lb. per sq. in. and on the 10° curve, 14 000 lb. per sq. in. Strangely the higher lateral bending stresses in the outer rail on both curves occur under the rear driver of each group except for the last group. As was stated in the discussion of the position of the flanges of this locomotive with respect to the rail on curved track and of the impressions given on copper wire, it is not clear how forces do act to change the direction of this locomotive at other than the first pair of drivers.

With Locomotive 10254 the lateral bending stresses in the inner rail are small (see Fig. 33), the greatest being 8 400 lb. per sq. in. on the 10° curve

and 6 100 lb. per sq. in. on the 6° curve. In the outer rail the lateral bending stresses are also low, the greatest at 50 miles per hour on the 6° curve being 9 600 lb. per sq. in. and at 40 miles per hour on the 10° curve 11 700 lb. per sq. in., both being under the first driver of the second half of the locomotive, which according to the observed position of the flanges is the principal guiding wheel for the second half of the locomotive. Other than for this driver, the lateral bending stresses in the outer rail are generally small and fairly uniform. The values of the lateral bending stresses, taken in connection with the position of the flanges of the wheels when the locomotive is at rest, already described, indicate that the first three outer wheels are the principal factors in changing direction in the first half of the locomotive and the first outer driver of the second half of the locomotive is the principal factor for the second half. Attention has already been called to the higher vertical bending stress under the last named driver at the higher speeds and to the consequent higher load transmitted to the rail by this driver.

The three forms of the Westinghouse-Baldwin locomotive showed much the same characteristics in the development of lateral bending stresses, and the magnitude of the stresses in general did not differ greatly. (See Figs. 34 to 36.) Locomotive 10302 (the original form) gave one exception to this statement. The lateral bending stress in the inner rail under the second driver at 5 miles per hour was extremely high, 22 200 lb. per sq. in., and this was accompanied by an extremely high vertical bending stress, 29 800 lb. per sq. in. Other than this value, the lateral bending stresses under the wheels of the three locomotives are fairly comparable at all speeds and in both rails of both curves. The higher values of the lateral bending stresses in the inner rail of the 10° curve range from 12 000 to 15 000 lb. per sq. in. and in the outer rail, from 10 000 to 12 000 lb. per sq. in.; in the inner rail of the 6° curve, from 10 000 to 12 000 lb. per sq. in., and in the outer rail, from 11 000 to 16 000 lb. per sq. in. In the outer rail of both curves at the highest speeds, the higher lateral bending stresses in general occur under the rear driver of the two groups, an unexpected result. The higher lateral bending stresses in the outer rail in general do not occur under those drivers that would be expected to participate strongly in changing the direction of the locomotives. In fact, the first outer driver of each half of the locomotive does not participate in the guiding action, which is taken principally by the outer truck wheels ahead of the drivers. The high lateral bending stresses in the outer rail under the rear outer driver of each group of drivers at the higher speeds on both curves may have some relation to the larger flange wear of these wheels reported by the railroad company, although it is difficult to understand why the flanges of these two drivers should be strongly pressed against the rail, and the tests with the copper wire gave no indication of such contact. It may be noted that at the highest speeds the vertical bending stress under these two drivers has also been increased greatly, indicating that the loads carried by these drivers are considerably higher than the average of those carried by the other drivers on the outer rail.

The lateral bending stresses under the wheels of the loaded freight cars on both curves (see Fig. 37) are significant. The stress in the outer rail of

both curves indicates that the flange of the front outer wheel of each truck acts to change the direction of the truck so as to traverse the curve. The lateral bending stresses in the outer rail on both curves at all speeds under all wheels almost without exception act to increase the curvature of the rail, and they increase with increase in speed. The lateral bending stresses in the inner rail on both curves almost without exception act to straighten the rail, and they decrease numerically with increase in speed. It will be seen that on the 10° curve at the speed corresponding to the super-elevation the magnitude of the lateral bending stress in the outer rail under any wheel is almost the same as that in the inner rail under the wheel on the same axle. The action is to spread both rails at the wheels and to narrow the gauge slightly at a point between wheels. Some of the lateral bending stresses are relatively high. The lateral bending stress of 9 500 lb. per sq. in. in the outer rail of the 10° curve under the first wheel of the rear truck of the second car is 60% of the corresponding vertical bending stress and 80% of the vertical bending stress under the same wheel on straight track at 40 miles per hour. In general, the lateral bending stresses are nearly as high as those found with Locomotive 10254—the locomotive which developed the lowest lateral bending stresses.

It is worthy of note that for all the electric locomotives the lateral bending of the outer rail on both curves at the highest speeds is generally strongly outward under every wheel with a few minor exceptions, a condition that differs markedly from that found with the drivers of all the steam locomotives with which tests have been made.

14.—Lateral Bending Moments.—It is of interest to represent the lateral bending action in the rail by the lateral bending moments calculated from the observed lateral bending stresses by the use of the lateral section modulus, $\frac{I}{c}$, of the rail, and to plot the values under and between wheels to show the distribution and the intensity along the locomotive and cars. As the position, nature, and stiffness of the support of the rail laterally on the tie is not known, the values of the lateral pressures against the rail can not be estimated. It is evident that the conditions of the lateral support of the rail may be quite different from that of its vertical support by the tie and that the elastic properties of the lateral support are unknown. The test values give, therefore, the only information on the lateral action of the rail.

Figs. 40 to 43 give diagrams of the lateral bending moments in the outer and inner rails for all the locomotives and for the loaded freight cars. As before, positive bending is taken to be that which increases the curvature of the rail, and negative bending that which tends to straighten the rail. The graphical representation may be helpful in easily getting some idea of the action of the several wheels and the effect of speed on lateral bending of the rail, and also in making comparisons of the behavior of the locomotives. It should be borne in mind that the lateral bending moments in the rail may be taken to be expressive of pressures and movements trans-

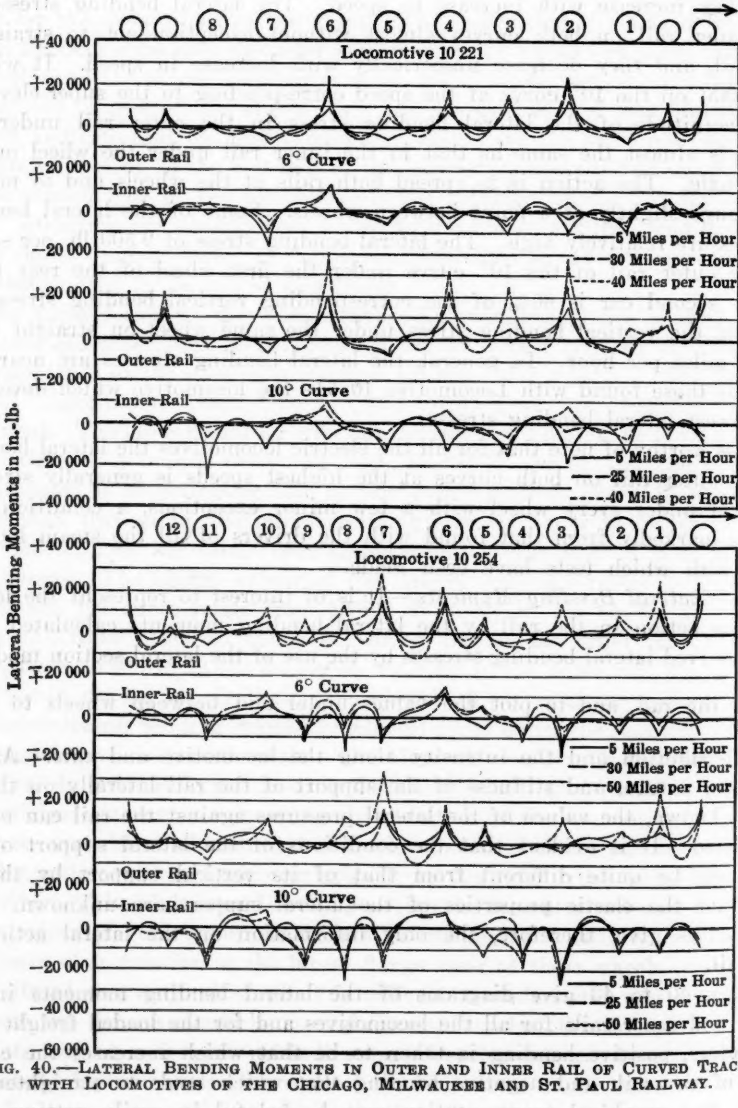


FIG. 40.—LATERAL BENDING MOMENTS IN OUTER AND INNER RAIL OF CURVED TRACK WITH LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

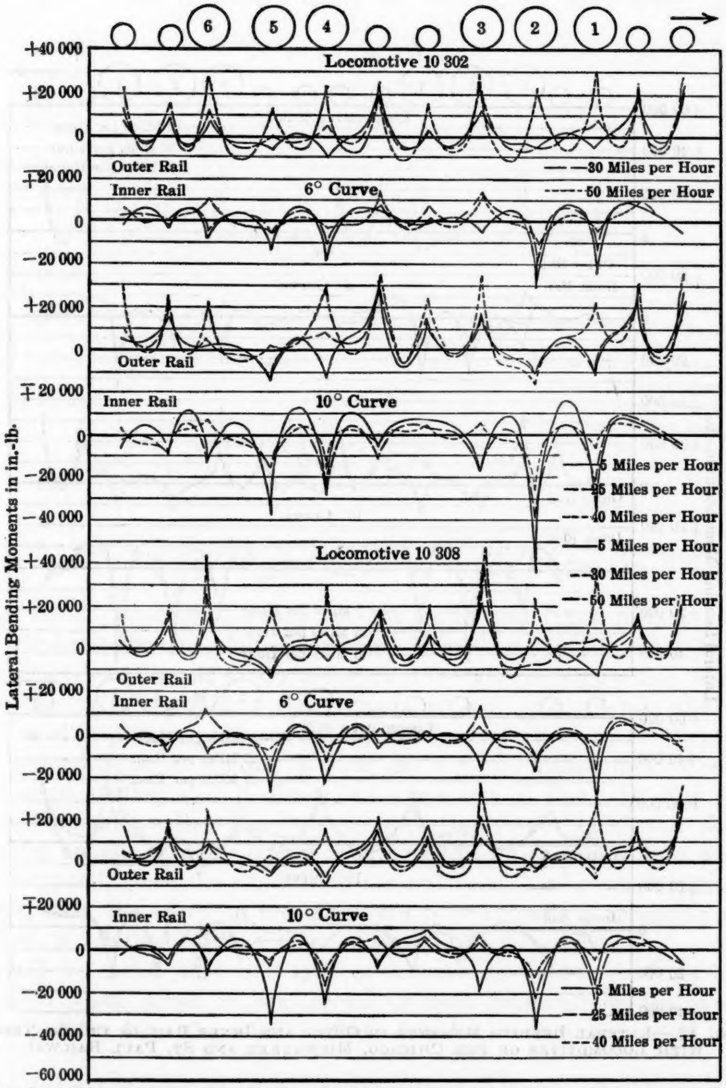


FIG. 41.—LATERAL BENDING MOMENTS IN OUTER AND INNER RAIL OF CURVED TRACK WITH LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

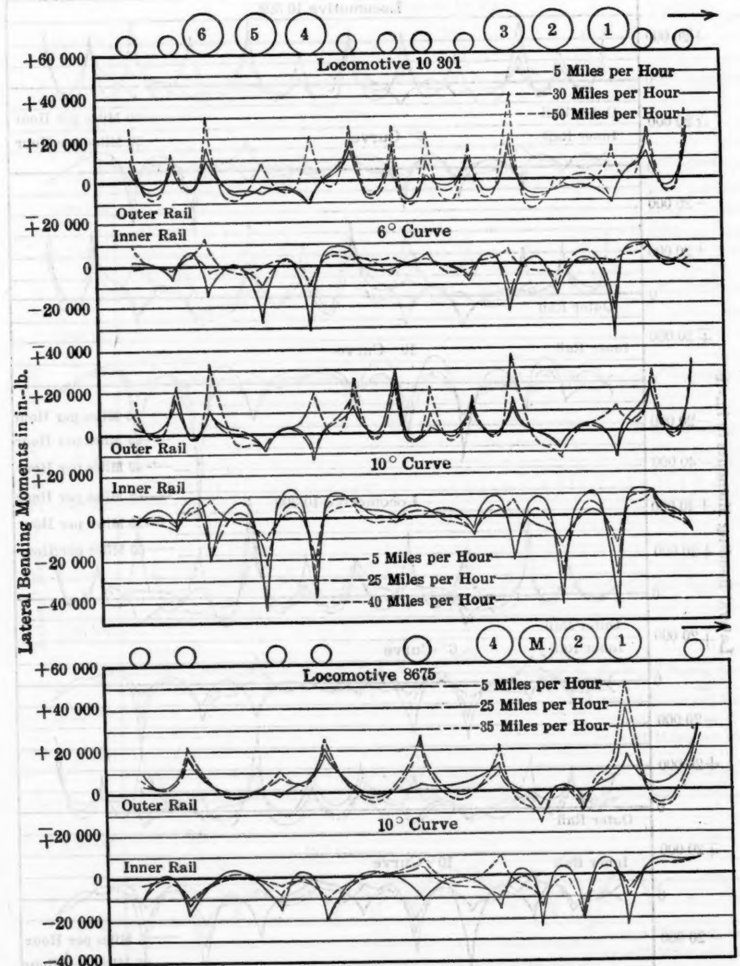


FIG. 42.—LATERAL BENDING MOMENTS IN OUTER AND INNER RAIL OF CURVED TRACK WITH LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

mitted to ties and ballast affecting the maintenance of track line and surface. The relations between the lateral bending for the several locomotives have already been discussed under Article 13, "Lateral Bending Stresses in Rail on Curved Track"; the diagrams of lateral bending moments bring out clearly many of the characteristics of the locomotives there described.

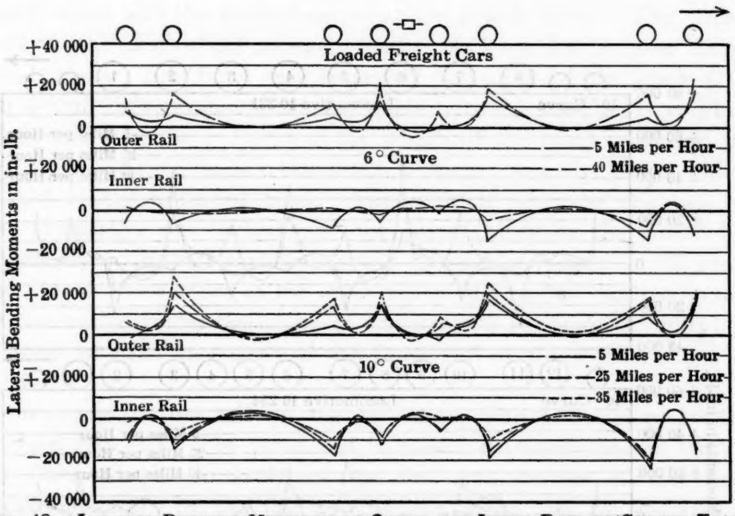


FIG. 43.—LATERAL BENDING MOMENTS IN OUTER AND INNER RAIL OF CURVED TRACK WITH LOADED FREIGHT CARS OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

The lateral bending moments for the outer and inner rails have been added algebraically and plotted for three locomotives in Fig. 44. The preceding diagrams (Figs. 40 to 43) brought out the large bending action developed in the two rails separately; these diagrams show the intense and severe lateral bending action produced in the track as a whole as the locomotive passes around a curve, tending to push the ties and ballast first to one side and then to another and to throw the curve out of line. In Fig. 45 the combined lateral bending moments for outer and inner rails show that for the loaded freight cars the lateral bending effect on the track as a whole is relatively small, since the moment in the two rails is of opposite sign. For the cars, then, the effect on the alignment is principally a spreading action and there is relatively little general distortion of the curve.

It will be noted that one type of locomotive produces effects on the track as a whole which in comparison to those produced on the two rails separately differ from the relative effects found with another type of locomotive.

15.—Lateral Movement of the Rails on Curved Track.—To obtain further information of conditions bearing on the maintenance of curved track, some measurements were made on the lateral movement of the head of the rail from its initial or original position as the locomotive passed around the curve at about 2 miles per hour. A bar and an attached Ames dial gauge were held between the rail and a fixed stake, 5 ft. away. The dial was read as

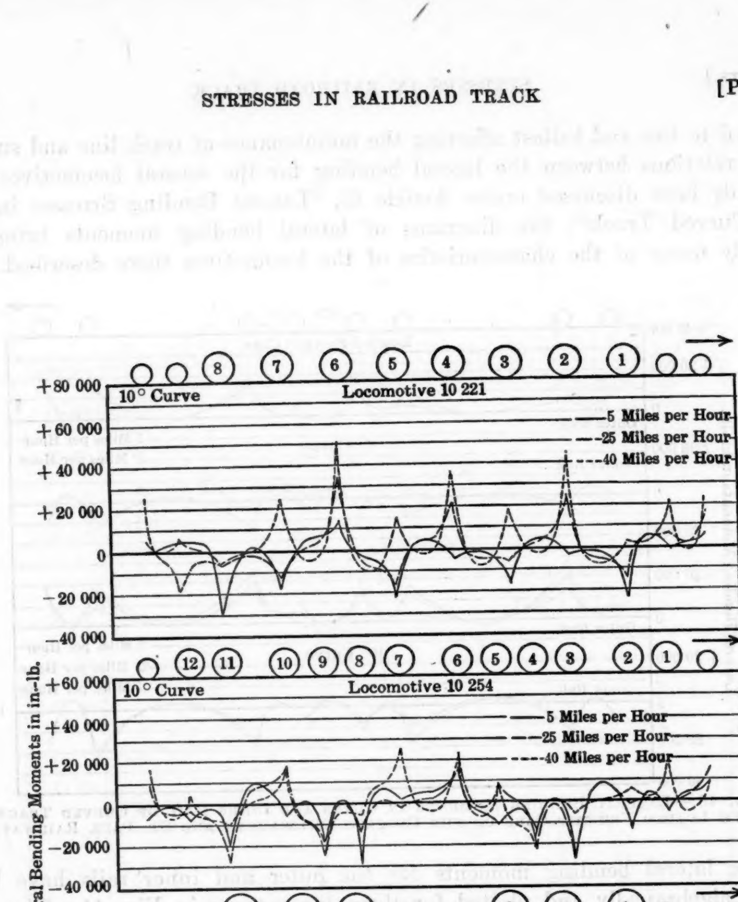


FIG. 44.—COMBINED LATERAL BENDING MOMENTS IN THE TWO RAILS OF THE 10° CURVE WITH LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

each wheel passed by. Any tilting of the rail will, of course, affect the lateral movement measured. Figs. 46 and 47 give some of the results of the tests. The values shown are generally averages of two tests.

With Locomotive 8675 (Mikado type) the outer rail of the 10° curve (see Fig. 46) is moved outwardly of the track more than 0.10 in. at the front truck wheel and the same distance at the fourth driver. The outward movement at the first wheel of each tender truck is also greater than at the second wheel. The inner rail is moved outwardly of the track at the first and second drivers about 0.22 in. This outward bending is continued by the other drivers, the trailer, and the wheels of the tender, ranging from 0.10 to 0.17 in. At all points the track is spread and the gauge widened beyond that of the unloaded track. The position taken by the two rails when the locomotive was backed over the curve (given by the dotted line) differs from that with the locomotives going forward principally in the greatly increased outward movement of the outer rail at the trailer (0.25 in.) (showing that the trailer guides the group of drivers, with apparent difficulty) and the increased outward movement of the inner rail at the second, third, and fourth drivers.

The movement of the rail on the 10° curve with Locomotive 10254 is peculiar (see Fig. 46). The outer rail is moved outwardly of the track by the first three wheels, inwardly by the first two drivers of the second group, and outwardly by the last two of this group and by the first one of the third group. The inner rail is moved outwardly throughout the length of the locomotive. The deflection is pronounced at the first driver of the second group and the second and third drivers of the third group, reaching 0.25 in. The widened gauge remains fairly constant for the whole distance. It is seen that the movement of the rails from their original position is in long swings as compared with the distances between the inflection points of the vertical and lateral bending stresses found in the rails themselves. The measurements taken when this locomotive was backed over the curve agreed very closely in magnitude and direction with the reversed position found with the locomotive going forward.

The outer rail of the 6° curve moved outward in the passage of Locomotive 10302, and the inner rail also for much of the length of the locomotive. Here, again, the increased gauge of the track remained fairly constant as the locomotive passed by. The record taken of the inner rail of the 10° curve shows a marked outward movement at the middle one of the first group of drivers—0.28 in.

The diagram given by Locomotive 10308 for the 10° curve shows an improvement over that given by Locomotive 10302. There is no large movement of either rail, and the gauge although widened remains nearly constant. Diagrams made with the locomotive backing are nearly identical with the reversed form of the forward ones.

The diagrams with Locomotive 10301 also are an improvement over those made by the original form of the locomotive. The widened gauge is nearly

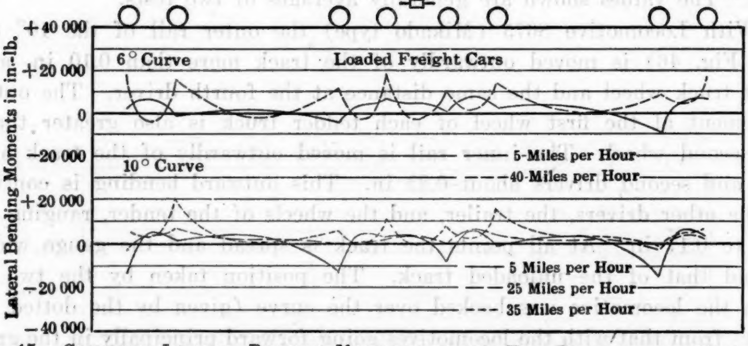


FIG. 45.—COMBINED LATERAL BENDING MOMENTS IN THE TWO RAILS OF THE 6° CURVE AND 10° CURVE WITH LOADED FREIGHT CARS OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

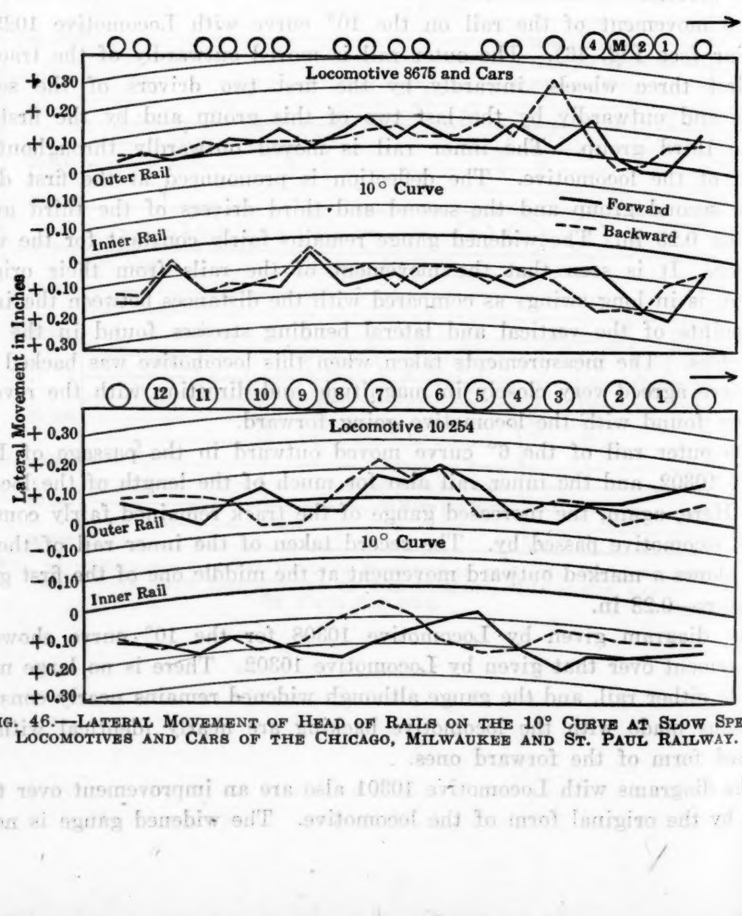


FIG. 46.—LATERAL MOVEMENT OF HEAD OF RAILS ON THE 10° CURVE AT SLOW SPEED. LOCOMOTIVES AND CARS OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

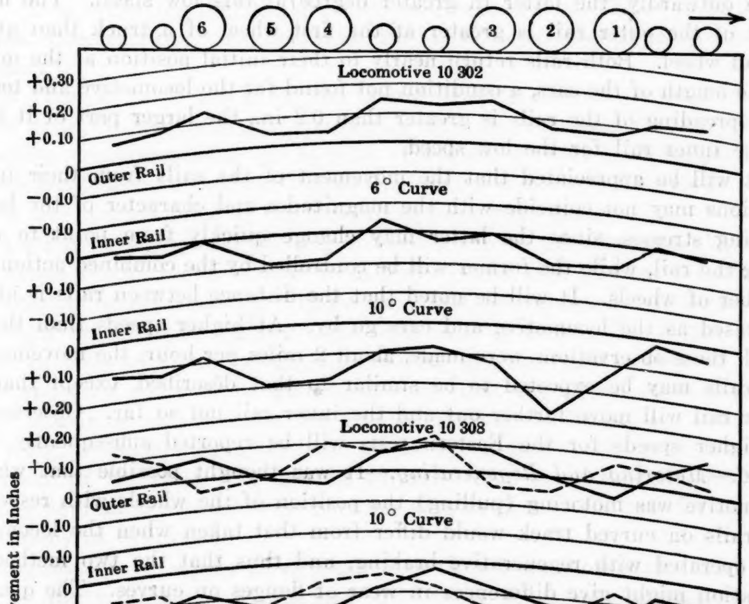


FIG. 47.—LATERAL MOVEMENT OF HEAD OF RAILS ON THE 6° AND 10° CURVES AT SLOW SPEED. LOCOMOTIVES OF THE CHICAGO, MILWAUKEE AND ST. PAUL RAILWAY.

constant. The movement of the rail with the locomotive backing was found to agree closely with that when it was going forward.

For the loaded freight cars (see Fig. 46), both outer and inner rails move outwardly, the latter in greater degree at this low speed. The movement of the outer rail is greater at the first wheel of a truck than at the second wheel. Both rails return nearly to their initial position at the middle of the length of the cars, a condition not found for the locomotive and tender. The spreading of the rails is greater than 0.2 in., the larger part of it being in the inner rail for the low speed.

It will be appreciated that the movement of the rails from their initial positions may not coincide with the magnitudes and character of the lateral bending stresses, since the latter may change quickly from point to point along the rail, while the former will be controlled by the combined action of a number of wheels. It will be noted that the distance between rails is always increased as the locomotive and cars go by. At higher speeds than that at which these observations were made, about 2 miles per hour, the movement of the rails may be expected to be similar to that described, except that the outer rail will move farther out and the inner rail not so far. Observations at higher speeds for the Eastern tests will be reported subsequently.

16.—Motoring and Regenerating.—It was thought possible that when a locomotive was motoring (pulling) the position of the wheels with respect to the rails on curved track would differ from that taken when the locomotive was operated with regenerative braking, and thus that the two methods of operation might give differences in wear of flanges on curves. The question was one of considerable interest to the railroad company. A possible source for such a difference lay in the direction of the friction between the drivers and the rail, the wheel tending to slip backward on the rail when the locomotive was motoring, and forward when it was regenerating. In traversing the curve the wheel on the outer rail must slip backward, or its companion on the inner rail forward, or possibly both. The wheel which slips longitudinally on the rail will thus be the one of the pair that has the best opportunity to slip laterally. It is seen then that the direction of possible longitudinal slip may determine conditions of lateral slip and thus also the position of the flanges with respect to the outer rail. As such changes would also modify the magnitude and direction of the lateral forces on the rail, it was thought well to make tests with two locomotives coupled together, one motoring and the other regenerating, and then reversing their operation and later their relative position. Any difference in the lateral bending of the rails might indicate differences in opportunities for flange wear during motoring and regenerating operation.

The tests were made on the 10° curve. The speed used was 25 miles per hour, approximately the speed corresponding to the super-elevation of the track, and also about the greatest speed at which regenerative braking is used with so high a retarding effort. Values of the braking and motoring tractive force developed during the regeneration tests were read from the characteristic curves for the two types of motors by means of values of the line voltage

and field and armature current observed in the locomotive cab as the test section was passed. The characteristic curves were furnished by L. Wiley, Assistant Electrical Engineer of the Chicago, Milwaukee and St. Paul Railway. It is estimated that the draw-bar pull ranged from 35 000 to 55 000 lb. in the various runs. As operated, the speed was maintained practically constant.

In Figs. 48 to 51 are given the lateral bending stresses in the outer and inner rails for one locomotive motoring and another regenerating, the one motoring sometimes being in the rear. Locomotive 10304 is an original form of the Westinghouse-Baldwin locomotives and is similar to Locomotive 10302 which was not available at the time these tests were made.

A study of the lateral bending stresses given in the diagrams shows that the values are generally similar in magnitude and sign whether a locomotive is motoring or regenerating. There are small differences, but these are such as might be found in making tests from time to time. The values already given in tests with the locomotives coasting are also very similar in magnitude and sign. It may be concluded, then, that the lateral stresses in both outer and inner rail do not differ greatly whether the locomotive is coasting, motoring, or regenerating, at least at the speed corresponding to the superelevation of the track. It may be expected that the flanges of the wheels occupy approximately the same position with respect to the rail for the three conditions of operation.

Impressions of copper wire, taken with these tests showed that with Locomotive 10221 the flange of the first outer truck wheel bore against the outer rail, and also the first outer wheel of the rear truck, the second wheel of each of these trucks being also close to the rail. This condition is true whether the locomotive was motoring or regenerating and whether it was in front of or behind another locomotive. It was also found that the flanges of the second inner driver of each group were usually close to the inner rail, even though the lateral bending stresses in the outer rail at the corresponding outer wheel indicated a large outward bending. From these impressions and the lateral bending stresses it is not clear how any but the first group of drivers is guided around the curve. It is evident that groups of two axles have an advantage in traversing curves.

Impressions of copper wire taken with Locomotive 10308 showed that the flange of the first outer truck wheel bore against the outer rail, as did the outer truck wheel just ahead of the second group of drivers and the first outer wheel of the rear truck, whether the locomotive was motoring or regenerating and the same condition was found when the locomotive was coasting over the curve. It is evident that the flanges of the two wheels named furnish the guiding action for the two groups of drivers. The same conditions were found with Locomotive 10304, but it will be noted that the lateral bending stress at the outer truck wheel ahead of the second group of drivers was greater than with Locomotive 10308, as was true in the test with Locomotive 10302 when coasting. In the tests with all the Westinghouse-Baldwin locomotives, at the speed of 25 miles per hour, usually the flange of the rear inner

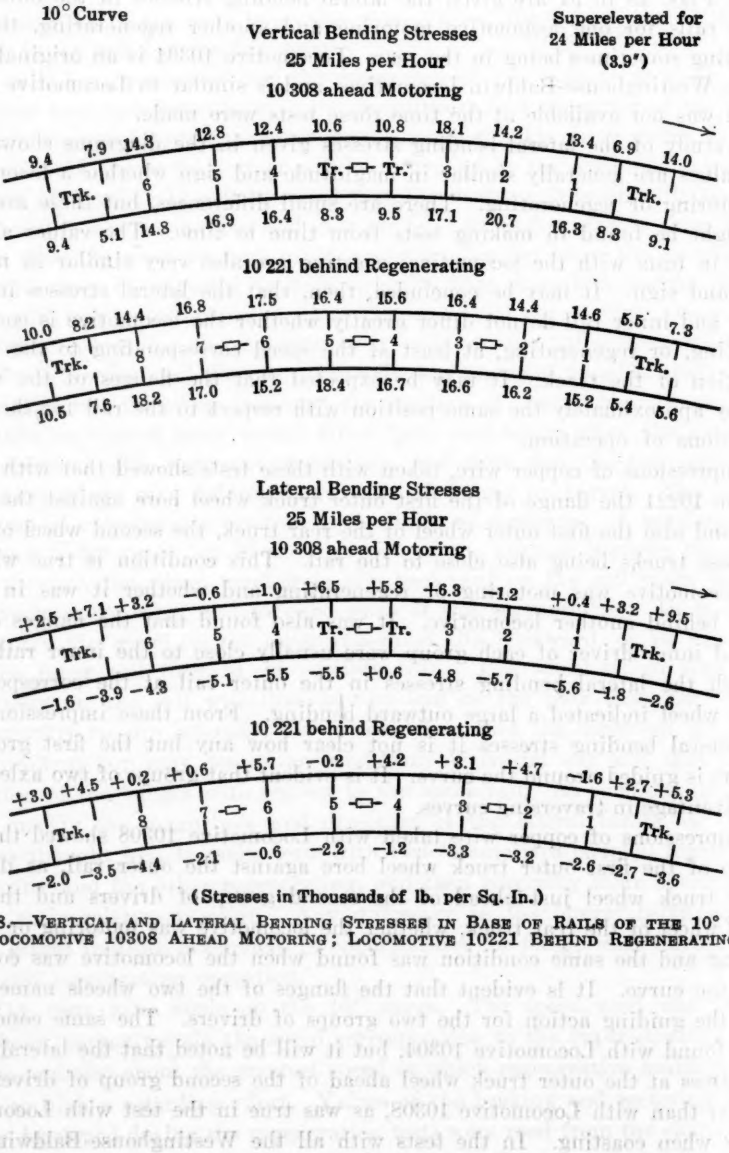


FIG. 48.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF THE 10° CURVE. LOCOMOTIVE 10308 AHEAD MOTORING; LOCOMOTIVE 10221 BEHIND REGENERATING.

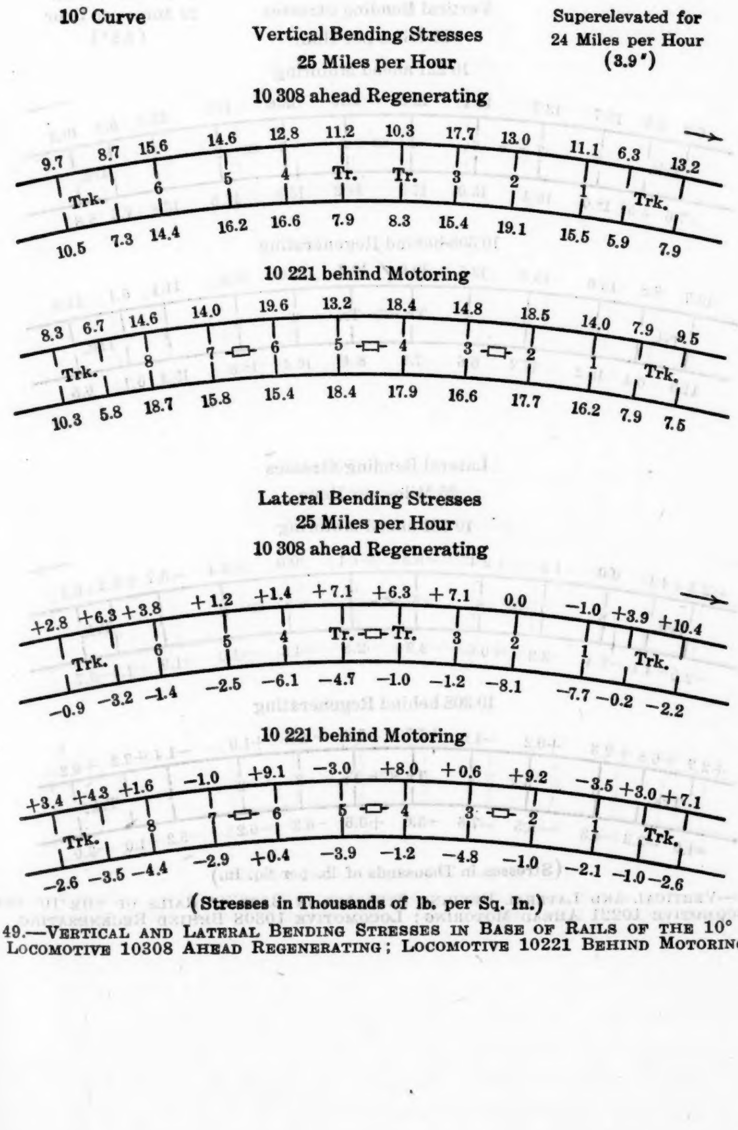


FIG. 49.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF THE 10° CURVE. LOCOMOTIVE 10308 AHEAD REGENERATING; LOCOMOTIVE 10221 BEHIND MOTORING.

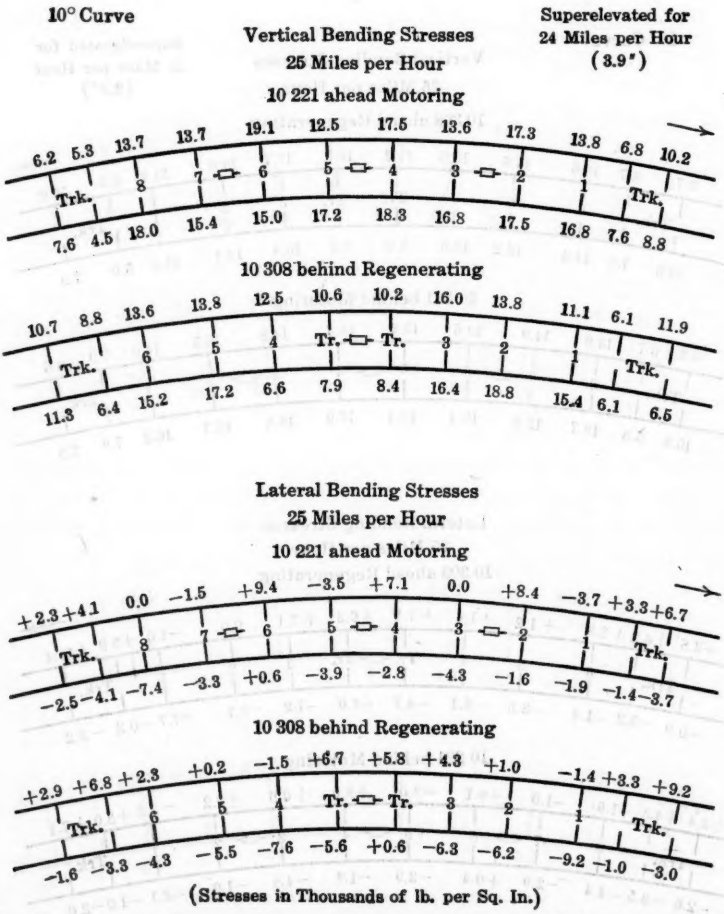


FIG. 50.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF THE 10° CURVE. LOCOMOTIVE 10221 AHEAD MOTORING; LOCOMOTIVE 10308 BEHIND REGENERATING.

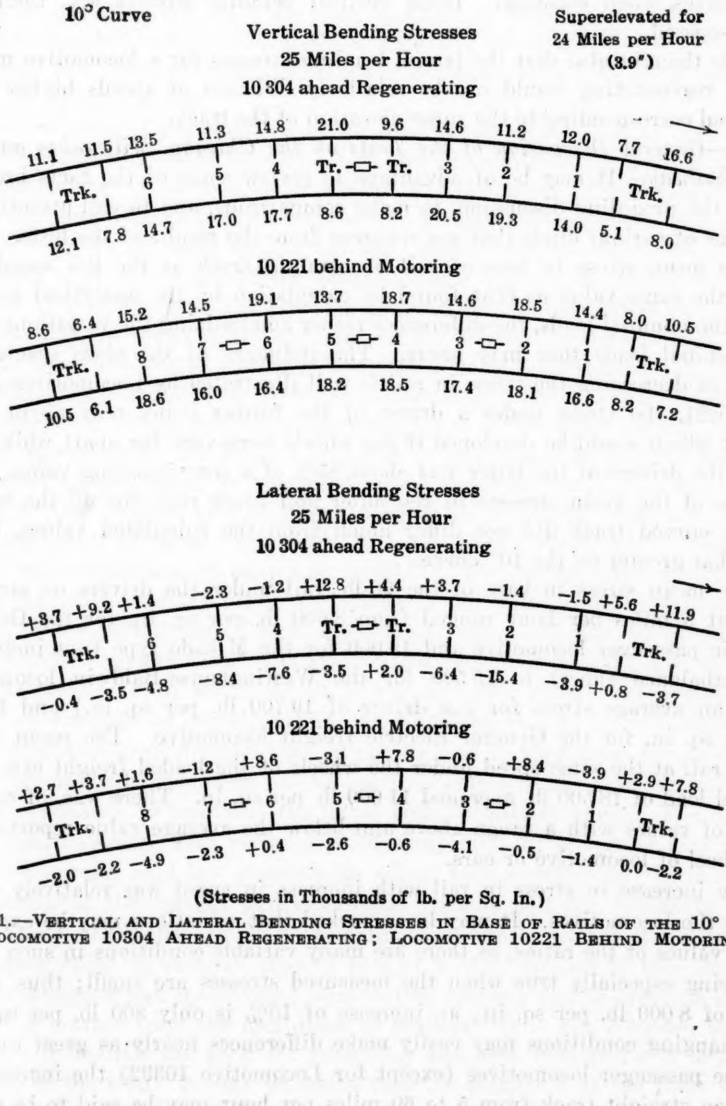


FIG. 51.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF THE 10° CURVE. LOCOMOTIVE 10304 AHEAD REGENERATING; LOCOMOTIVE 10221 BEHIND MOTORING.

driver of each group, and, in some cases, those of both the second and the rear driver, were close to the inner rail, while at the same time the companion driver produced outward bending in the outer rail.

It was also found that the vertical bending stresses developed in the motoring-regenerating tests (see Figs. 48 to 51) did not differ greatly with the two methods of operating and were much the same as those found with the same locomotives when coasting. These vertical bending stresses are, therefore, not discussed.

It is thought also that the lateral bending stresses for a locomotive motoring or regenerating would not be relatively different at speeds higher than the speed corresponding to the super-elevation of the track.

17.—General Discussion of the Tests on the Chicago, Milwaukee and St. Paul Railway.—It may be of advantage to review some of the facts brought out in the preceding discussion, to make comparisons, and to call attention to relations of various kinds that are apparent from the results of the tests.

The mean stress in base of rail on straight track at the low speed had much the same value as that found by calculation by the analytical method using the nominal loads, the differences rather emphasizing the variations from the nominal loads that may occur. The influence of the close spacing of wheels in decreasing the stress in rail is well illustrated by Locomotives 10254 and 10221, the stress under a driver of the former being only about 50% of that which would be developed if the wheels were very far apart while that under the drivers of the latter was about 85% of a corresponding value. The average of the mean stresses in the outer and inner rails for all the wheels on the curved track did not differ much from the calculated values, being somewhat greater on the 10° curve.

The mean stress in base of the 90-lb. rail under the drivers on straight track at 5 miles per hour ranged from 8 000 lb. per sq. in. for the General Electric passenger locomotive and 12 000 for the Mikado type (not including counterbalance effect) to 15 500 for the Westinghouse-Baldwin locomotive (with an average stress for one driver of 19 700 lb. per sq. in.) and 17 000 lb. per sq. in. for the General Electric freight locomotive. The mean stress in the rail at the same speed under the wheels of the loaded freight cars with a wheel load of 18 500 lb. averaged 11 000 lb. per sq. in. There was, of course, a belt of values with a range above and below the average value reported for any wheel of locomotive or cars.

The increase in stress in rail with increase in speed was relatively small for all the locomotives. It may be remarked that variations may be expected in the values of the ratios, as there are many variable conditions in such tests, this being especially true when the measured stresses are small; thus, for a stress of 8 000 lb. per sq. in., an increase of 10% is only 800 lb. per sq. in., and changing conditions may easily make differences nearly as great as this. For the passenger locomotives (except for Locomotive 10302) the increase in stress on straight track from 5 to 60 miles per hour may be said to be about 12% (see Fig. 14). The corresponding increase in stress was about 1 500 lb. per sq. in. The speed effect from 5 to 40 miles per hour was relatively smaller.

That of the General Electric freight locomotive was not large. That of the Mikado locomotive was larger, but was only 10%; this does not include the effect of counterbalance. On the 6° curve the speed effect differed but little from the value for straight track. The 10° curve gave a somewhat higher value, but the difference is not marked. The values for the loaded freight cars on straight track and 10° curve are greater, but for these the increase in mean stress is only 13% and 25%, respectively. All these values of the effect of speed are smaller than were obtained in former tests. It should be borne in mind that the values noted are all average values; in all the locomotives and cars there are increases under individual wheels that are much larger than the average given.

The relation of the speed effect for dead or unsprung load to that carried on effective springs is of interest, as is also the relation of diameter of wheel to speed effect. It is frequently stated that the effect of speed on track is inversely proportional to the square of the diameter of the wheel and directly proportional to the dead weight carried. Sometimes, the first power of the diameter is given instead of the square. Table 1 gives the unsprung weight carried by one driver of the General Electric freight locomotive as 8 100 lb., equal to 29% of the average total load on the driver, the corresponding values for the General Electric passenger locomotive being 4 800 lb. and 24%, and for the Westinghouse-Baldwin passenger locomotive, 3 900 lb. and 12 per cent. The diameters of the drivers of the three types of locomotive are 52, 44, and 68 in., respectively. In Table 4 and Fig. 14 are given the average values of the speed effects found, as measured by the mean stress in base of rail. A study of these values and of the other data of the tests* does not indicate that the effect of speed on bending stresses in rail varies with diameter of wheel and proportion of unsprung weight to the extent given by the principles quoted previously. The variations in the values due to a number of causes do not permit the effect of diameter of wheel and amount of unsprung weight to be segregated, but it is evident that their effects, although important, are far smaller than would result if the principle cited were sound. The effect of diameter of wheel, within the limits of size and speed considered, from analytical considerations would not seem to be very great for good track, whatever might be the effect on the bearing strains in the rail (detrusion) or in the resistance to traction—the latter being relatively small. The assumption that the speed effect varies with the proportion of unsprung load implies that the sprung load has a negligible effect, thus overlooking the defects of the action of the springs and equalizing systems and the flexibility of the track itself. Doubtless, too, a load applied between the wheels of a pair exerts less speed effect on one rail than if half of it were applied to the one wheel. Poorly designed and maintained equalization systems and inadequate springs, too, may detract from the usefulness of springs. It is not intended by these remarks to belittle the need for providing adequate springs for a large proportion of the load. It may be added that the condition of the track surface

* In Locomotive 8675 the unsprung weights for the first, second, third, and fourth drivers are 20, 22, 33, and 20% of the wheel loads, respectively. The diameter of the drivers is 63 in. The increase in stress in rail under the drivers with change from 5 to 40 miles per hour on straight track is about 9%, not counting the effect of counterbalance.

must have an important influence on the speed effect; uneven track, that is, track with a succession of hard spots and soft spots may be expected to give large speed effects. Attention should also be called to the advantage of having adequate easement curves when the proportion of unsprung weight is large, particularly when this weight is low, as when it is carried directly on the axle.

The foregoing discussion relates to the average values of the mean stress in base of rail. Due to the lateral bending of the rail, the ratio of stress at outside edge to mean stress will vary. On straight track (see Tables 6 and 7), the average of these ratios for the various wheels of the several locomotives ranged from 0.82 to 1.33, the first value being under the trailer of the Mikado locomotive and the second under a driver of a Westinghouse-Baldwin passenger locomotive, the lateral bending stress in the second instance having a value of 4 500 lb. per sq. in. The ratios were generally different for the two sides of the locomotive, sometimes markedly different, and the ratio at one wheel differed from that at another. Generally, there were one or more wheels in each locomotive that gave a markedly higher outward lateral bending of the rail than the others, the reason for the higher stresses not being apparent.

Also, for a given wheel, the individual observations of lateral bending stresses in the rail of straight track varied widely from the average stress corresponding to the average ratio just considered. Lateral bending stresses as great as 9 000 lb. per sq. in. were found in connection with a mean stress or vertical bending stress of 22 000 lb. per sq. in., making a stress of 31 000 lb. per sq. in. at the outside edge, though there were few of this magnitude. It is evident, of course, that these lateral stresses and the accompanying lateral movement of the rail are factors which produce stresses and movements in ties and ballast and thus affect track maintenance and also enter into the maintenance requirements of the locomotives. It may be said, however, that the lateral bending stresses and lateral movements found in the tests with these locomotives are, in general, less than have been found in some of the tests of straight track made previously. With the electric locomotives the absence of stress due to counterbalance was especially noticeable.

It is to be expected that with the great diversity in wheel loads, wheel spacing, wheel grouping, and articulation of running gears and devices for regulating flexibility, the four types of locomotives in traversing the curved track will show a variety in the magnitude and distribution of the lateral bending stresses in rail, in the division among the wheels of the work of guiding the locomotive around the curve, and in effecting the lateral slip of the wheels on the rail which is an essential element in negotiating a curve. Differences may be expected in the flexibility of the locomotives, both as a whole and in their various parts, and also in the effect on the alignment of the track and its necessary maintenance. A discussion of the effect of the details of design of the different locomotives and their influence on the track as shown by the tests cannot well be taken up at this time. The mechanical departments of the railroads and the builders of locomotives have long given attention to the requirements of locomotives for traversing curved track. The results herein

reported may be expected to be of service in the further improvement of details of this part of locomotive design. Some of the methods and results may be suggestive of ways for securing information bearing on the effect and the usefulness of devices and of methods of correcting defects which may be found to exist.* The data reported here will give opportunity for the study of various questions which have been raised concerning certain features of locomotive design. It is to be expected that conditions giving unusual or unexpected stresses in the rail will produce similar effects in the locomotive structure.

The tests show that the lateral bending stresses in the rail on curved track, and, therefore, the lateral pressures on the rail, are relatively very great. For the curves used (6° and 10°) relatively large lateral pressures on the rail must be accepted as a necessity; the problem is how by proper design and maintenance to keep the values as small as possible and to make the track structure as resistant as is feasible.

All the locomotives behaved well in traversing the curved track even at the highest speed run, although, of course, differences were noticeable. The speeds of 50 miles per hour on the 6° curve and 40 miles per hour on the 10° curve run by the passenger locomotives were rather high, in view of the super-elevation used, but the locomotives ran the curves very smoothly except where the alignment was not good. The speed corresponding to the super-elevation of the track was only three-fifths of the maximum speeds run, but the tests do not indicate that the maximum lateral bending stresses were unduly increased by reason of this difference.

The effect of imperfect alignment in parts of the curved track on the behavior of the locomotives, as in the vicinity of the pile bridge on the 10° curve, and also the effect of an easement curve that was imperfectly maintained serve to emphasize the importance of maintaining excellence in line and surface if trains are to be operated at high speeds around such curves. At the higher speeds it appears also that easement curves of adequate length are very useful in adjusting the tracks and coupling connections to take the curve under the best conditions and thus to reduce the wear of both locomotive and track.

The tests have again brought out the inter-relation of quality of maintenance of both track and rolling stock to effects of traffic on the maintenance of both and thus emphasize the value of keeping up a high standard of excellence in the maintenance of track and of locomotives and cars. High excellence in the make-up of the track structure and in the design of locomotives and cars likewise contributes to low maintenance costs.

* An excellent illustration of improvements made by changes in the design of locomotives after track tests had been made, whereby a critical excessive stress developed in the rail of a 10° curve was decreased by one-third, is given in *Circular No. D. V.-344* of the Mechanical Division of the American Railway Association, a paper on "The Relation of Track Stresses to Locomotive Design" by C. T. Ripley, Chief Mechanical Engineer of the Atchison, Topeka and Santa Fé Railway. The paper brings out in an excellent way the intimate relation between locomotive design and stresses in track.

III.—Tests on Eastern Railroads

18.—The Phenomena.—The tests on Eastern railroads were undertaken mainly with the view of obtaining information on the effect of canting the rail inwardly of the track, upon the stresses in the rail, and upon other matters affecting track maintenance, as compared with the results found with upright rail. The tests also gave information on the action of track laid with the heavy rail used on three of the railroads.

Whether there are advantages in so supporting the rail that its axis is inclined inwardly of the track instead of being vertical has been a mooted question. The advocates of canting have held that by adzing the ties to give the right slope or by using tie-plates with the outer end thicker than the inner one (called inclined tie-plates), the center of the bearing of the tread of the wheels of locomotives and cars will be more nearly on the center of the head of the rail and the wear of the rail will extend over a greater width of the head, thus increasing the life of the rail, and that the load will be more directly transmitted to the tie than is the case when the axis of the rail is vertical. It is further said that canting tends to prevent widening of the gauge under traffic and otherwise reduces the cost of maintenance. The inclined tie-plate also usually projects farther on one side of the rail than on the other, the greater thickness at the side having the greater projection may present an economic advantage in design. The advocates of a vertical position for the rail feel that better results will be obtained with a flat tie-plate proportioned so that it projects outwardly from the rail a distance sufficiently greater than its inward projection that the resultant of the bearing force of wheel on rail will pass through the center of the tie-plates; that the condition of the tread of wheels is so diverse and the conditions of track and rolling stock affecting the direction and magnitudes of the lateral pressures so variable that it is impracticable so to cant the rail as to give the advantages claimed; that it is difficult to provide a construction near and through turn-outs that will be substantial and workmanlike without undue expense; and that dissatisfaction with vertical rail may generally be laid to neglected maintenance and to lack of tie-plates or to small and poorly proportioned tie-plates.

On curved track the effect of canting the rail may be expected to be more complicated than on straight track, and the conditions may differ for inner and outer rail and under the several wheels of the locomotive and cars. It is evident then that tests and observations will be needed on both straight and curved track and that separate consideration should be given to the two kinds of track.

As the test party had had no previous experience with the problem, all the tests had to be made in the ways that seemed to give the best promise of results, and the utility of the methods could not be determined until after the data were reduced and assembled. It had been judged that the lateral bending stresses in the rail, as measured by the stremmatograph, would furnish the most useful information obtainable and that measurements of the lateral movement of the rail and of its tilting as the load passed by would serve well as an

auxiliary means of obtaining information. The lateral bending stresses were found to give information of value bearing on the canting of the rail and other special conditions of the track, but it is now seen that a much larger number of measurements of lateral movement and lateral tilting of the rail at several points on the track with a better and more systematic development of methods would have furnished information more directly applicable to the problems than was appreciated in advance.

It is apparent that the phenomena of lateral movement, lateral tilting, and lateral bending of the rail are intimately connected not only with the condition of the tread of the wheel and the position of the rail with respect to the vertical, but also with the amount of eccentricity of the tie-plate, the nature of the wood of the tie and its elastic properties, the condition of the ballast support, and the lateral movements and forces of the locomotive and cars. These elements are bound together in an intricate way. To find the influence of a single one would be a difficult task. To help to understand the action of these factors a further statement of the conditions and factors present when wheel loads pass over the track may be useful. Some of them may seem minor matters and others indefinite—all of them will be subject to considerable variation. The conditions to be found on straight track will be taken up first:

(*a*) The gravity load transmitted by wheels to rails on track that is level transversely constitutes a vertical force, which, of course, has no horizontal or lateral component.

(*b*) Lateral movements of the load due to variations in track and oscillations and nosings of locomotive and cars produce lateral forces acting against the rail, either inwardly or outwardly of the track. These forces will be applied to the rail by friction between the tread of the wheel and the head of the rail, and less frequently by pressure from the flange of the wheel itself.

(*c*) Lateral changes in the direction of motion of the wheels of a frame or truck from time to time, whether by lateral slipping or otherwise, will develop lateral forces on the rail that are of consequence in contributing to the variability of results.

(*d*) It has been shown in previous progress reports that the ordinary coned wheel running along an upright rail develops a lateral bending in the rail. For perfect conditions of wheel, rail, and tie, it would appear that this outward thrust may be connected with the differences in distances traveled by the wheel tread at the two edges of the surface of contact, due to the differences in the circumferences of the wheel at these two points and also possibly to an unsymmetrical area of contact of wheel on rail when the center of bearing is at one side of the middle of the head. This lateral bending of the rail persists even after the wheels have come to rest. It would be of interest to learn whether rail that is so canted as to bring the center of bearing at the middle of the head would develop lateral bending of the rail to the same extent.

(*e*) The resultant of the vertical load and the lateral force may not pass through the mid-point of the base of the rail. The use of an eccentric

tie-plate, that is, one having a greater projection on one side of the rail than on the other, is, of course, based on the belief that the resultant is not vertical, and, since the longer projection is placed on the outside, that the resultant inclines outwardly. As the position of the resultant will be variable, the design of the tie-plate should be based upon the average position of this force.

(*f*) For other positions of the resultant the distribution of pressure on the tie will not be uniform from end to end of the plate. If e is the relative eccentricity of the resultant of the applied forces, given in terms of the length of the tie-plate (for example, $\frac{1}{10}$ when the resultant passes 1 in. from the middle of a tie-plate 10 in. long), and p_0 is the average pressure of tie-plate on the tie, it is easy to show that the intensity of pressure, p, at an end of the tie-plate will be given by the equation:

$$p = (1 \pm 6\,e)\,p_0 \qquad (34)^*$$

when the relative eccentricity is less than $\frac{1}{6}$ and,

$$p = \frac{2}{3}\,\frac{p_0}{\frac{1}{2} - e} \qquad (35)$$

at the outside end when the relative eccentricity is greater than $\frac{1}{6}$. In Fig. 52 three distributions of the bearing pressure are shown. For uniform pressure the resultant of the applied forces will pass through the mid-point of the plate. If the resultant of the applied forces cuts the bottom of the plate one-sixth of its length away from its mid-point $\left(\text{relative eccentricity, } \frac{1}{6}\right)$, the pressure at one edge will be double the average and that at the other will be zero. If the resultant passes through a point one-fourth of the length from the mid-point, the pressure at one edge will be two and two-thirds times that when uniformly distributed over the tie-plate, and zero at and beyond the quarter-point of the plate. In the latter case the resultant would pass near or outside the edge of the base of rail. In this connection a distinction should be made between the eccentricity of the resultant of the applied forces with respect to the tie-plate and the eccentricity of an unsymmetrical tie-plate, the latter being used as the distance between the mid-point of the length of the tie-plate and the axis of the rail as laid.

(*g*) When the resultant passes to one side of the mid-point of the tie-plate, the tie is unevenly compressed along the length of the plate. As the modulus of elasticity of wood for bearing pressure is relatively small (less than one-fiftieth part of that for tension and compression lengthwise of the fibers, counting the effect of both detrusion and compression), there may be a relatively large difference in the depth that the plate depresses the tie at the two

* This equation number follows the last number used in the third progress report, *Transactions*, Am. Soc. C. E., Vol. LXXXVI (1923), p. 980.

ends of the plate. As a result, the rail will tilt under the load, the head of the rail will move inward or outward, and the gauge will narrow or widen. Besides, if the intensity of bearing pressure is greater than the elastic bearing limit of the wood, repetitions of such applications will result in cutting the tie and give permanent changes in gauge and alignment. It is evident that the length of tie-plate and its eccentricity will have an important influence upon the action of the rail and tie under load and thus, also, upon matters of track maintenance. This tilting and lateral movement may be quite independent of the lateral bending of the rail due to the lateral forces already referred to.

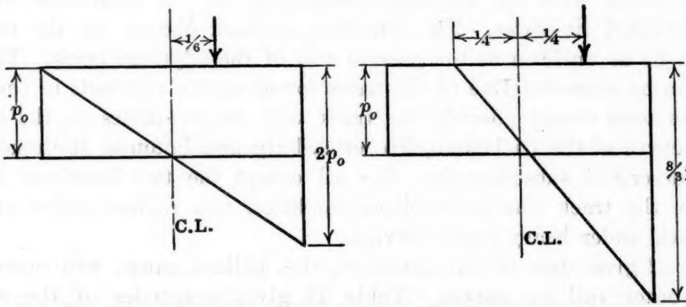


Fig. 52.—Distribution of Pressure Over Tie-Plates for Different Positions of Resultant.

(*h*) The flexural position of the tie at the rail may also affect the inclination of the rail, although not to the same extent as the tie compression. If the tie at this point slopes outwardly when under load, as in the case of a centerbound tie, the rail will tilt outwardly; if the slope is inward, as with an endbound tie, the rail will tilt inwardly.

(*i*) The order of magnitude of the depressions, slopes, tilting, and lateral movement may not be large, and yet the effect on track maintenance may be of importance.

(*j*) It is apparent that with all the variations of locomotives and cars and of track conditions great variations will be found in the effect on rail and tie. One rail may differ considerably from the other, and the conditions will vary from point to point along the track. Both average results and ranges in values should be considered and a study made of the meaning of the variations.

On curved track the conditions are further complicated. The transverse inclination of the track gives inclination to both rails, changing the inclination of the line joining the middle of the head of the rail and the middle of the length of the tie-plate, in effect increasing the eccentricity of the tie-plate of the outer rail for speeds below the speed corresponding to the super-elevation and decreasing it for speeds above it, and having the opposite effect for the inner rail. As was shown in the third progress report, the wheels of a fixed frame must be made to change direction in traversing the curve; the flange of the leading outer wheel, or of a wheel ahead of the group, by bearing against the outer rail acts to cause lateral slippage of two or more of the leading wheels

of the frame and thus develops an important lateral force outwardly on the rail. By the turning action other outward or inward lateral forces in both rails are developed, which cause important lateral bending stresses in the rail. These various lateral forces and the combined effect of centrifugal force and super-elevation may be expected to influence the effect of canting the rail and of eccentricity of tie-plate.

19.—The Track.—The tests were made on track in the condition found at the time. No special work had been done before the test except that on the Baltimore and Ohio Railroad inclined tie-plates had been replaced ten days before on a stretch of straight track and on a 7° curve in order that tests might be made with flat tie-plates comparable to the conditions of track having inclined tie-plates. The stretches of track chosen for the tests appeared to be as uniform as the general run of the adjacent track. They are thought to be representative of the track found on the railroads in the vicinity of the tests except possibly the track with flat tie-plates on the tangent and 2½° curve of the Richmond, Fredericksburg and Potomac Railroad which will be described subsequently. For all except the two locations just referred to, the track was in excellent condition and representative of high-grade track under heavy traffic service.

Table 12 gives data of rail, tie-plates, ties, ballast, gauge, and super-elevation of outer rail on curves. Table 13 gives properties of the sections of the rails where tests were made and Fig. 53 the rail sections. Fig. 54 shows the design of the tie-plates. All inclined tie-plates had a slope of 1 in 20.

The average cant of the rails at the test locations, both straight track and curved track, is given in Table 14. The cant was found by placing a straight bar across the track under the rails and measuring the deviation of the base of rail from this bar, no load being on the track. The value thus found varied from point to point along the track, regardless of the kind of tie-plate used and the condition of the track, and the values in Table 14 are the averages of five observations on each rail. In several cases, the rail was found to be canted outwardly of the track, one such instance being on straight track.

The tests on the Baltimore and Ohio Railroad were conducted on the east-bound track at Harpers Ferry, W. Va. On the straight track the rail was worn only slightly. On the two 7° curves the rail was in good condition, except that on the 7° curve having flat tie-plates, the head of the inner rail had been mashed down considerably, this condition existing before the tie-plates were changed. The alignment and surface were good. The two curves had a high super-elevation (6.7 and 7.1 in.), and the greatest speed used, 40 miles per hour, was only slightly above that corresponding to the speed of super-elevation. With the heavy traffic over this track, the tests were conducted with difficulty.

The tests on the railroad of the Reading Company (formerly the Philadelphia and Reading Railway) were conducted on east-bound track at and near Myerstown, Pa. From Myerstown west there are four tracks and a few

TABLE 12.—DATA OF THE TRACK.

Track.	Tie-plates.	Rail section.	Year of rolling.	Ties, in feet and inches.	Number of ties, in 33 ft.	Depth of ballast, in inches.	Gauge, in feet and inches.	Super-elevation, in inches.	Speed corresponding to super-elevation, in miles per hour.
Baltimore & Ohio R. R.:									
Straight	Flat	130-lb. R.E.	1921	Treated oak, 7 by 9 by 8 ft. 6 in.	18	18, Rock	4 ft. 8 3/4 in.	...	..
Straight	Inclined	"	"	"	"	"	4 " 8 9/16 "	...	..
7° curve	Flat	"	"	"	"	"	4 " 8 9/16 "	6.7	38
7° curve	Inclined	"	"	"	"	"	4 " 9 1/16 "	7.1	39
Reading Co.:									
Straight	Flat	"	1923	Treated oak, 7 by 9 by 8 ft. 6 in.	18	12, Rock	4 " 8 7/16 "	...	..
Straight	Inclined	"	"	"	"	"	4 " 8 5/8 "	...	..
Lehigh Valley R. R.:									
Straight	Inclined	136-lb. L. V.	1916	Treated oak, 7 by 9 by 8 ft. 6 in.	20	18, Rock	4 " 8 9/16 "	...	..
10° curve	Inclined	"	1923	"	"	"	4 " 8 5/8 "	6 1/4	31
Richmond, Fredericksburg & Potomac R. R.:									
Straight	Flat	100-lb. Am. Soc. C. E.	1914	Oak, 7 by 9 by 8 ft. 6 in.	19	Gravel	4 " 8 9/16 "	...	..
Straight	Inclined	100-lb. R. E.	1922	"	18	"	4 " 8 5/16 "	...	..
2 1/8° curve	Flat	100-lb. Am. Soc. C. E.	1914	"	21	"	4 " 8 11/16 "	4.0	49
5° curve	Inclined	100-lb. R. E.	1921	"	18	"	4 " 9 1/8 "	5.4	40

miles eastward there are three, so that the traffic on the test stretch is extremely heavy. The rail had been laid two months before. A few of the inclined tie-plates were loose. The ties were large and in good condition. The track was in general good condition.

TABLE 13.—PROPERTIES OF SECTIONS OF RAILS ON TEST TRACK.

Rail section (full section).	Area, in square inches.	Moment of Inertia.		Section Modulus.		
		For horizontal axis.	For vertical axis.	Horizontal Axis.		Vertical Axis.
				Base.	Head.	Base.
90-lb. A. R. A.-A........	8.80	38.7	7.5	15.2	12.6	2.9
100-lb. Am. Soc. C. E...	9.84	44.0	9.8	16.1	14.5	3.4
100-lb. R. E..............	9.95	49.0	9.4	17.8	15.1	3.5
130-lb. R. E..............	12.71	77.4	14.5	25.6	20.8	4.8
136-lb. L. V..............	13.35	86.6	17.7	28.3	22.0	5.4

The tests on the Lehigh Valley Railroad were conducted on east-bound track, 6 miles west of Mauch Chunk, Pa. The traffic was heavy. The tests were made on straight track and 10° curve (the latter being called the Ox Bow Curve), both having inclined tie-plates. The rail was in good condition, though that on the 10° curve, as shown in Fig. 101, was considerably worn even though the outer rail had been laid only two months before, the inner rail having been taken from the outer rail at the same time where it had been in service about six months. The tie-plates had been in the track and on the same ties since 1916. The surface and alignment were good, and the ballast was unusually good. On the 10° curve the super-elevation ($6\frac{1}{4}$ in.) corresponded to a speed of 31 miles per hour, and a speed of 35 miles per hour was used in the tests.

The tests on the 5° curve on the Richmond, Fredericksburg and Potomac Railroad were made north of Cherry Hill, Va., on track with inclined tie-plates, and near Brooke, Va., on straight track laid with flat and with inclined tie-plates and on a $2\frac{1}{2}$° curve with flat tie-plates. The 5° curve was in excellent condition, except that the alignment was only fair. The super-elevation was 5.4 in., corresponding to a speed of 40 miles per hour. The straight track having canted tie-plates was in excellent condition. The track having flat tie-plates, both the straight track and the $2\frac{1}{2}$° curve, was not in as good condition; it was planned soon to relay these stretches with new rail and inclined tie-plates. At the test location of straight track with flat tie-plates, near the beginning of the $2\frac{1}{2}$° curve, the right rail was 2 in. lower than the left, the location being chosen as the only straight track with flat tie-plates available. All tests were made on south-bound track.

Measurements of track depression were not made except in an incomplete manner, but it is thought that the modulus of elasticity of rail support, u, was about 2 000 on the Baltimore and Ohio Railroad, the Reading Company and the Lehigh Valley Railroad. The track structure was apparently both solid and stiff.

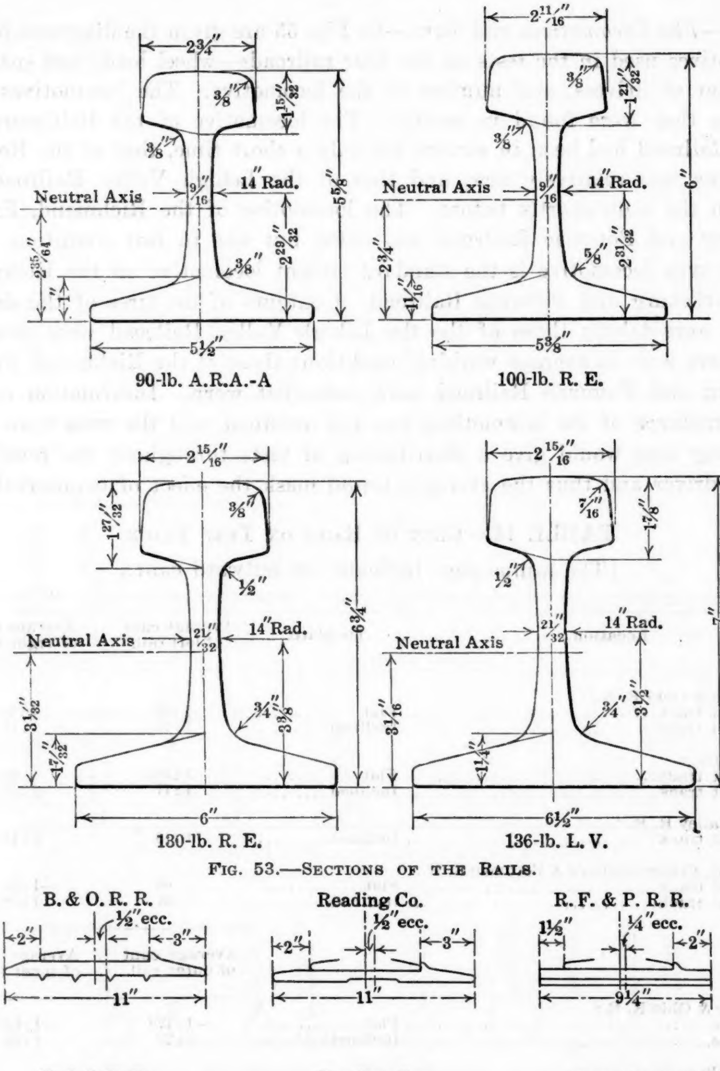


FIG. 53.—SECTIONS OF THE RAILS.

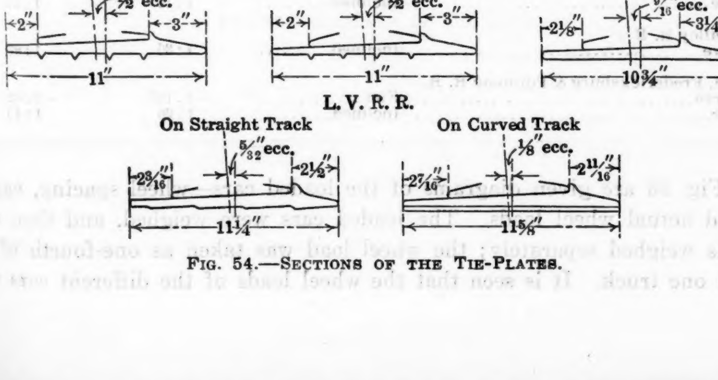


FIG. 54.—SECTIONS OF THE TIE-PLATES.

20.—The Locomotives and Cars.—In Fig. 55 are given the diagrams for the locomotives used in the tests on the four railroads—wheel loads and spacings, diameter of drivers, and number of the locomotive. The locomotives were used as they were found in service. The locomotive of the Baltimore and Ohio Railroad had been in service for only a short time, that of the Reading Company was relatively new, and that of the Lehigh Valley Railroad had been in the shop shortly before. The locomotive of the Richmond, Fredericksburg and Potomac Railroad was older, but was in fair condition. The Pacific type locomotive is the standard freight locomotive on the Richmond, Fredericksburg and Potomac Railroad. Contours of the tires of the driving wheels were taken; those of the the Lehigh Valley Railroad were new and the others were in average working condition; those of the Richmond, Fredericksburg and Potomac Railroad were somewhat worn. Information on the counterbalance of the locomotives was not obtained, and the runs were made in a way that would give a distribution of tests throughout the revolution of the driver and thus the averages would mask the effect of counterbalance.

TABLE 14.—Cant of Rails on Test Track.

(The minus sign indicates an outward cant.)

Location.	Tie-plate.	Average cant of left rail.	Average cant of right rail.
Baltimore & Ohio R. R.:			
Straight track	Flat	1:67	1:54
Straight track	Inclined	1:15	1:17
Reading Co.:			
Straight track	Flat	1:100	00
Straight track	Inclined	1:17	1:16
Lehigh Valley R. R.:			
Straight track	Inclined	1:18	1:18
Richmond, Fredericksburg & Potomac R. R.:			
Straight track	Flat	00	−1:30
Straight track	Inclined	1:21	1:18
		Average cant of outer rail.	**Average cant of inner rail.**
Baltimore & Ohio R. R.:			
7° Curve	Flat	−1:120	−1:120
7° Curve	Inclined	1:23	1:23
Reading Co.:			
1° Curve	Flat	1:60	−1:30
1° Curve	Inclined	1:15	1:24
Lehigh Valley R. R.:			
10° Curve	Inclined	1:36	1:27
Richmond, Fredericksburg & Potomac R. R.:			
2½° Curve	Flat	−1:115	−1:32
5° Curve	Inclined	1:26	1:43

In Fig. 56 are given diagrams of the loaded cars—wheel spacing, capacities, and actual wheel loads. The loaded cars were weighed, and then each half was weighed separately; the wheel load was taken as one-fourth of the load on one truck. It is seen that the wheel loads of the different cars vary

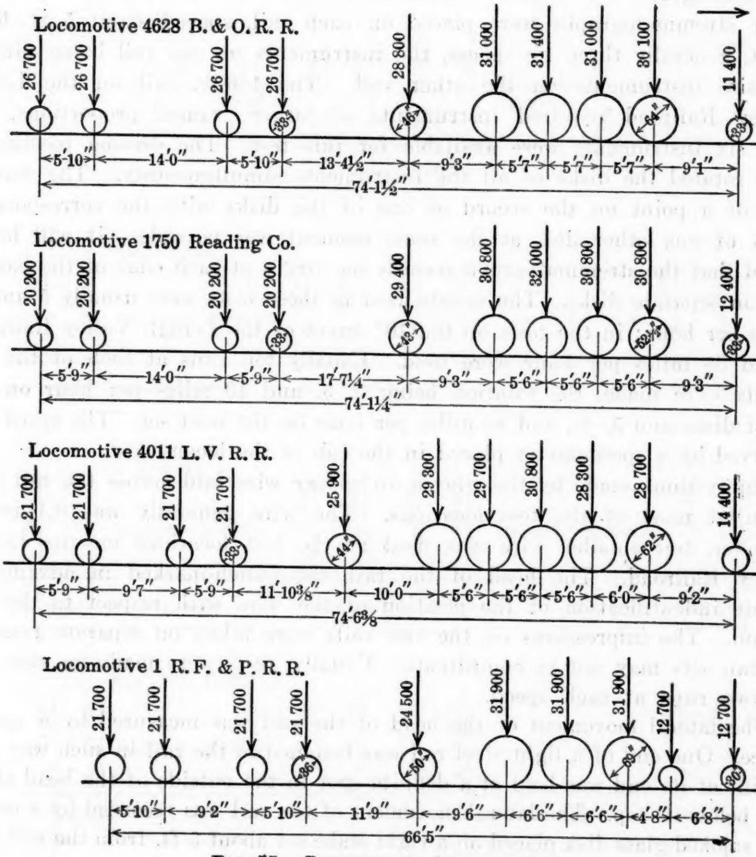


FIG. 55.—DIAGRAMS OF LOCOMOTIVES.

from 17 500 to 26 500 lb. The cars had been in service for some years; they had the variations that are to be found in cars in service. The wheels were not badly worn. The diameter of the wheels was 33 in.

It should be noted that the standard coning of the wheels of locomotives and cars of the Baltimore and Ohio Railroad is 1 in 13; that of all the other railroads is 1 in 20.

21.—Conduct of Tests and Reduction of Data.—The method of using the stremmatograph was similar to that described in the former progress reports. Four stremmatographs were placed on each rail, spaced about 5 ft. 6 in. apart, generally three tie spaces, the instruments on one rail being directly opposite instruments on the other rail. The 136-lb. rail on the Lehigh Valley Railroad required instruments of larger framed proportions, and only six instruments were available for this test. The driving mechanism used rotated the disks of all the instruments simultaneously. The correlation of a point on the record of one of the disks with the corresponding point of any other disk at the same moment was possible. It will be recalled that the stremmatograph records the strain at each edge of the base of rail on separate disks. The speeds used in these tests were usually 5 and 40 miles per hour; in the tests on the 10° curve of the Lehigh Valley Railroad, 5 and 35 miles per hour were used. Usually ten runs at each of the two speeds were made, the rotation being 5, 5, and 40 miles per hour on one set of disks and 5, 40, and 40 miles per hour on the next set. The speed was observed by a speedometer placed in the cab of the locomotive.

Impressions made by the wheels on copper wire laid across the rail were taken at most of the test locations. The wire generally was 0.1 in. in diameter, but smaller wire was used at the test locations on the Lehigh Valley Railroad. The head of the rail was punch-marked in advance to permit indentification of the position of the wire with respect to the rail section. The impressions on the two rails were taken on separate runs, so the two sets may not be co-ordinate. Usually, tests were made on each rail for two runs at each speed.

The lateral movement of the head of the rail was measured by a special device. One end of a light steel rod was fastened to the rail in such way that a point at its end was held at a definite spot in the outside of the head about ½ in. below its top. The lateral movement of the rail was recorded by a needle on a smoked-glass disk placed on a rigid stake set about 5 ft. from the rail, and this disk was rotated as the train passed by.

The tilting of the rail with reference to its initial position was measured by another device. A bar placed under the two rails was fastened so that the rail could pivot about one of its edges. A needle-bar connected to the head of the rail at a definite place about ½ in. below the top of the rail, held a needle which made a record on a smoked-glass disk, the disk being fastened to a support at the end of the bar and rotated by an outside mechanism. The apparatus permitted vertical and lateral movement of the rail and measured only the movement due to tilting of the rail.

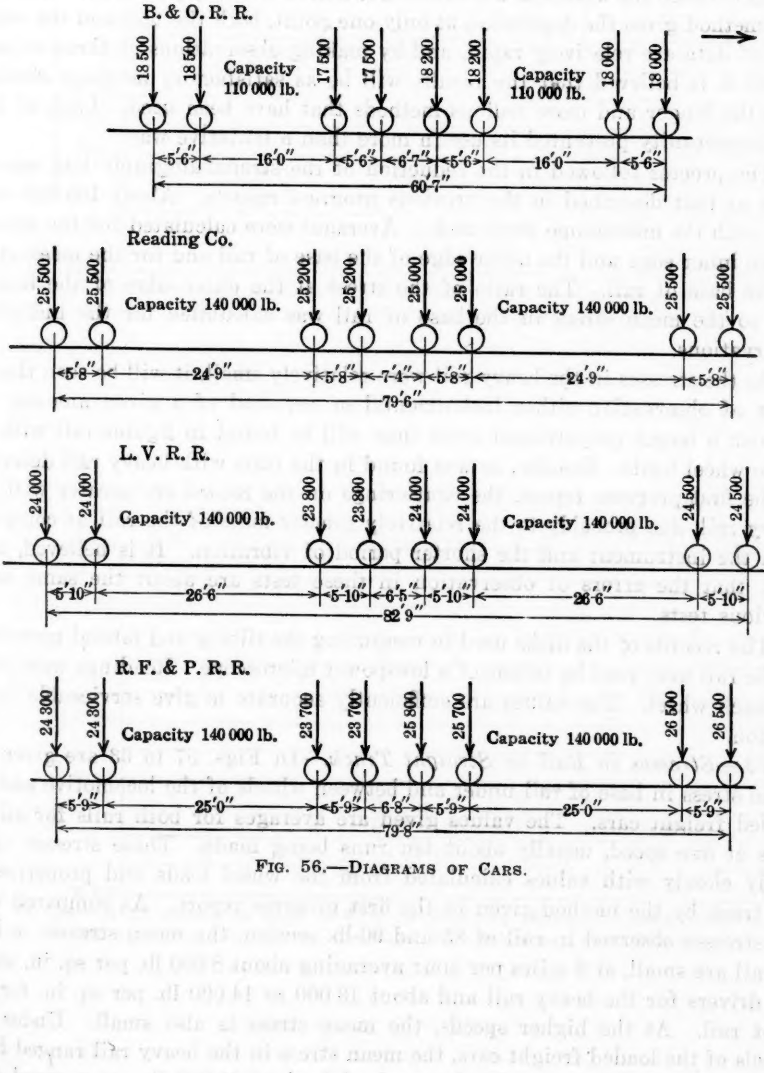


FIG. 56.—DIAGRAMS OF CARS.

The methods used on curved track were the same as on straight track. Care was taken to choose a location that was representative of the curve with respect to alignment, super-elevation, cant, and gauge.

A method for determining the track depressions under load was developed. One end of a wooden bar was fastened to the base of the rail with a hinged connection. The other end was pivoted at the top of a stake 10 ft. from the rail. At a point two-thirds of the distance out from the rail an Ames dial gauge was placed between the bar and another stake. The Ames dial was read at and between the wheels as the locomotive and cars slowly passed by. Although this method gives the depression at only one point, both the test and the reduction of data are relatively rapid, and by making observations at three or more points it is believed that the results will be as satisfactory as those obtained with the longer and more tedious methods that have been used. Lack of time and opportunity prevented its use in more than a tentative way.

The process followed in the reduction of the stremmatograph data was the same as that described in the previous progress reports. About 135 000 readings with the microscope were made. Averages were calculated for the stresses at the inner edge and the outer edge of the base of rail and for the mean stress in the base of rail. The ratio of the stress at the outer edge of the base of rail to the mean stress in the base of rail was calculated for the individual observations.

As the stresses in the heavy rail were relatively small, it will be seen that an error of observation either instrumental or personal of a given amount will furnish a larger proportional error than will be found in lighter rail with the same wheel loads. Besides, as was found in the tests with heavy rail described in the first progress report, the chatterings on the record are greater with the heavy rail, due probably to the relatively greater mass of the rail as compared with the instrument and the shorter period of vibration. It is believed, however, that the errors of observation in these tests are about the same as in previous tests.

The records of the disks used in measuring the tilting and lateral movement of the rail were read by means of a low-power microscope. Readings were made for each wheel. The values are sufficiently accurate to give serviceable information.

22.—Stresses in Rail on Straight Track.—In Figs. 57 to 63 are given the mean stress in base of rail under and between wheels of the locomotive and the loaded freight cars. The values given are averages for both rails for all the runs at one speed, usually about ten runs being made. These stresses agree fairly closely with values calculated from the wheel loads and properties of the track by the method given in the first progress report. As compared with the stresses observed in rail of 85 and 90-lb. section, the mean stresses in base of rail are small, at 5 miles per hour averaging about 8 000 lb. per sq. in. under the drivers for the heavy rail and about 13 000 or 14 000 lb. per sq. in. for the light rail. At the higher speeds, the mean stress is also small. Under the wheels of the loaded freight cars, the mean stress in the heavy rail ranged from 6 000 to 10 000 lb. per sq. in. for a wheel load of 25 000 lb., at a speed of 5 miles per hour, nearly as great as that found under the drivers of the locomo-

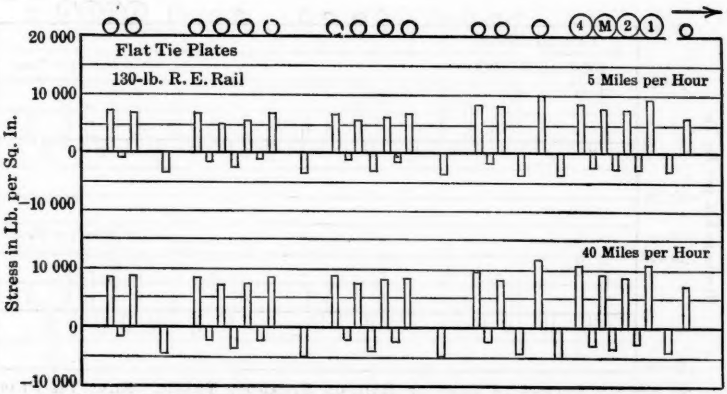


FIG. 57.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, BALTIMORE AND OHIO RAILROAD.

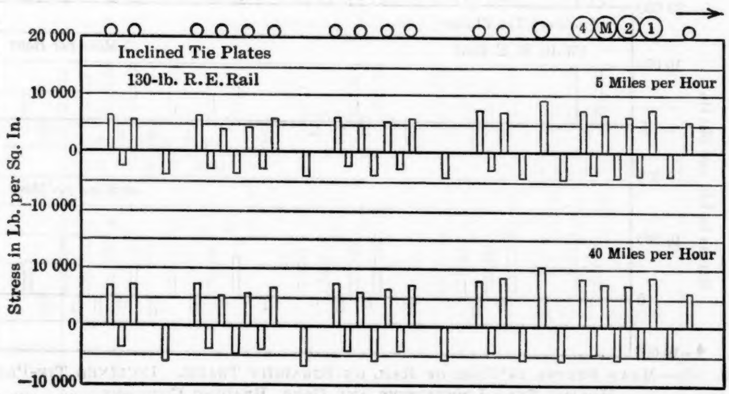


FIG. 58.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, BALTIMORE AND OHIO RAILROAD.

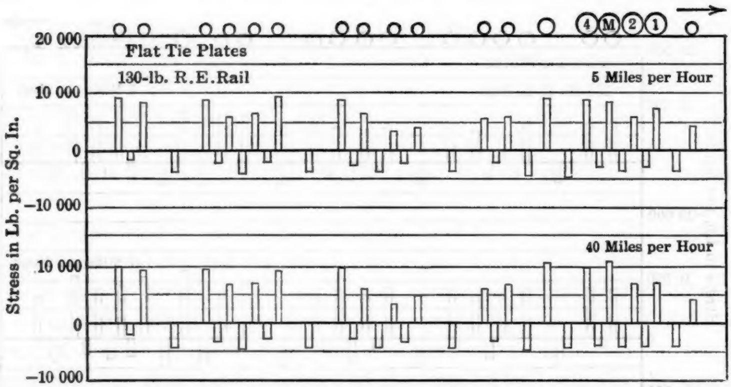


FIG. 59.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, READING COMPANY.

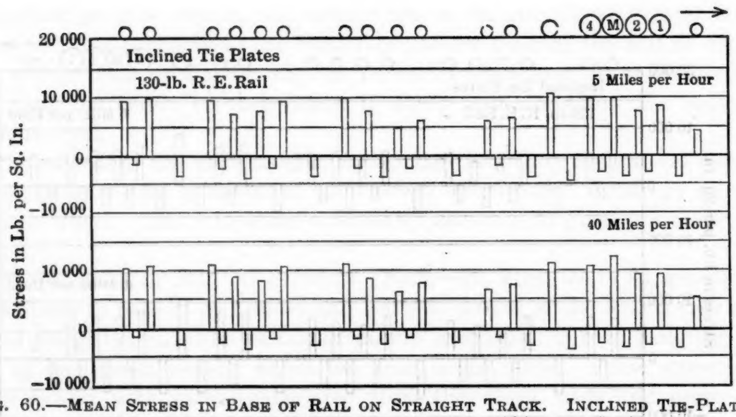


FIG. 60.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, READING COMPANY.

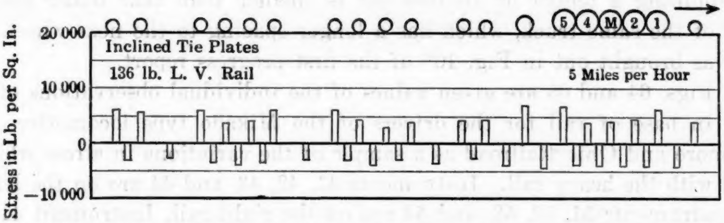


FIG. 61.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. INCLINED TIE-PLATES. SANTA FÉ TYPE LOCOMOTIVE AND CARS, LEHIGH VALLEY RAILROAD.

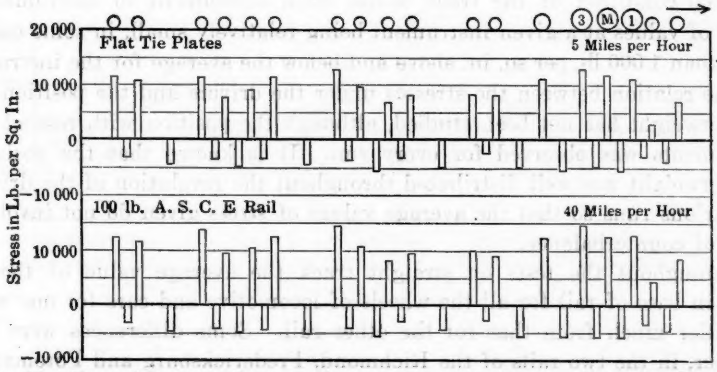


FIG. 62.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. FLAT TIE-PLATES. PACIFIC TYPE LOCOMOTIVE AND CARS, RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD.

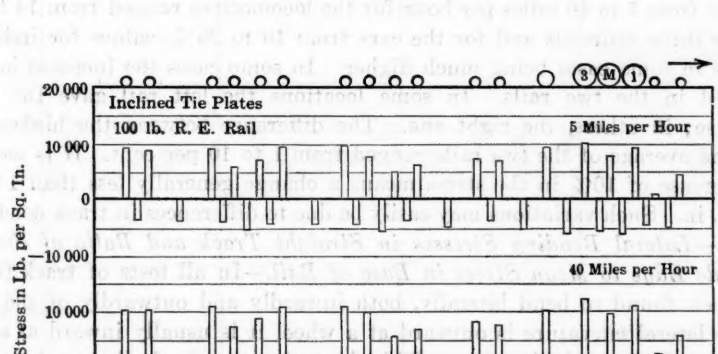


FIG. 63.—MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK. INCLINED TIE-PLATES. PACIFIC TYPE LOCOMOTIVE AND CARS, RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD.

tive. It is seen from the figures that the stress under a wheel at the end of a car adjoining a tender or another car is smaller than that under the other wheel of the same truck, which has a longer spacing to the next wheel. This fact was brought out in Fig. 10* of the first progress report.

In Figs. 64 and 65 are given values of the individual observations of mean stress in base of rail for the drivers of the Mikado type locomotive on the Baltimore and Ohio Railroad as a sample of the variations in stress which are found with the heavy rail. Instruments 41, 42, 43, and 44 are on the left rail, and Instruments 51, 52, 53, and 54 are on the right rail, Instrument 41 being opposite Instrument 51, etc. The main belt of values seems to extend about 3 500 lb. per sq. in. on each side of the average line. The variation of values on the whole is similar to that found in tests previously made. It is evident that the condition of the track varies from instrument to instrument, the range of values at a given instrument being relatively small, in some cases not more than 1 000 lb. per sq. in. above and below the average for the instruments.

The relation between the stresses under the drivers and the position of the counterweight has not been studied, although the position with respect to the instruments was observed for every run. It is known that the position of counterweight was well distributed throughout the revolution of the drivers in the various runs so that the average values of stress given do not involve any effect of counterbalance.

Throughout the tests on straight track the average value of the mean stress in base of rail for all the wheels of locomotive and cars for one rail did not differ much from that for the other rail. Some differences were found, however, in the two rails of the Richmond, Fredericksburg and Potomac Railroad, due in part in one case to one rail being lower than the other.

The average increase in mean stress in base of rail on straight track for a change from 5 to 40 miles per hour for the locomotives ranged from 14 to 17% on the three railroads and for the cars from 10 to 25%, values for individual wheels in some cases being much higher. In some cases the increase in stress differed in the two rails. In some locations the left rail gave the higher increase; in others, the right one. The difference between the higher value and the average of the two rails ranged from 1 to 10 per cent. It is seen that an increase of 10% in the stress means a change generally less than 1 000 lb. per sq. in. Such variations may easily be due to differences in track conditions.

23.—Lateral Bending Stresses in Straight Track and Ratio of Stress at Outside Edge to Mean Stress in Base of Rail.—In all tests of track the rail has been found to bend laterally, both inwardly and outwardly of the track. If the lateral curvature is outward at a wheel it is usually inward at a point between wheels, and *vice versa.* This flexural action is similar to the vertical bending of the rail, convex downward at a wheel and upward between wheels. The lateral bending of the rail in the sense here used should be distinguished from the larger outward or inward movement of the rail as a whole or for distances greater than, say, 40 in. for a group of wheels, in just the same way that the bending of a rail in the vertical plane must be distinguished from the

* *Transactions,* Am. Soc. C. E., Vol. LXXXII (1918), p. 1214.

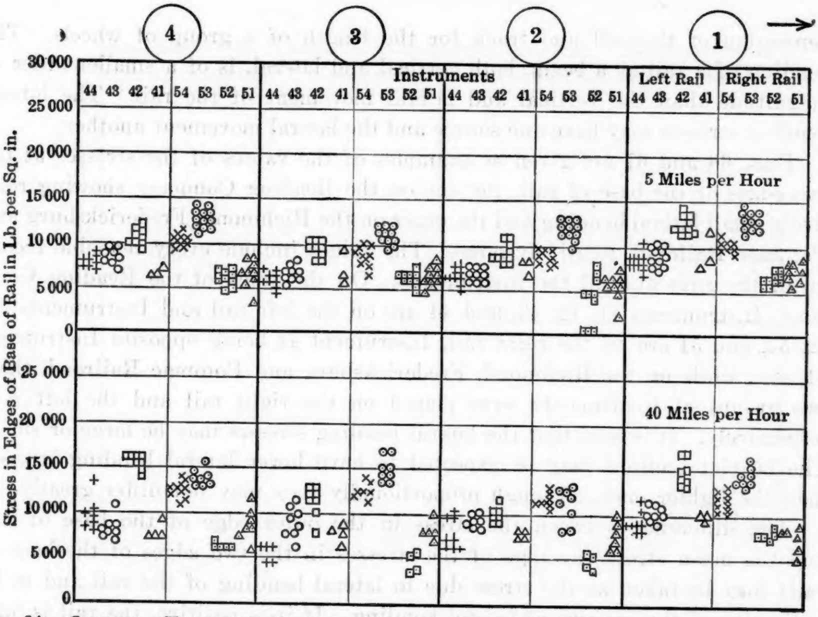


FIG. 64.—OBSERVED VALUES OF MEAN STRESS IN BASE OF RAILS AT THE EIGHT INSTRUMENTS ON STRAIGHT TRACK UNDER THE DRIVERS OF THE MIKADO TYPE LOCOMOTIVE OF THE BALTIMORE AND OHIO RAILROAD. FLAT TIE-PLATES.

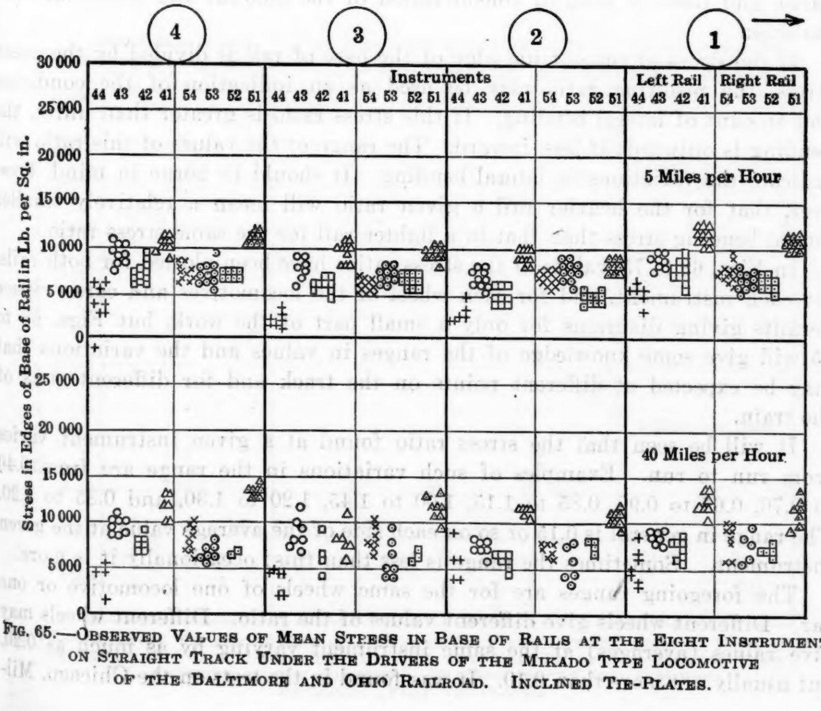


FIG. 65.—OBSERVED VALUES OF MEAN STRESS IN BASE OF RAILS AT THE EIGHT INSTRUMENTS ON STRAIGHT TRACK UNDER THE DRIVERS OF THE MIKADO TYPE LOCOMOTIVE OF THE BALTIMORE AND OHIO RAILROAD. INCLINED TIE-PLATES.

depression of the rail and track for the length of a group of wheels. The bending of a rail as a beam, both vertical and lateral, is of a smaller order of magnitude than the vertical and lateral movement of the rail. The lateral bending stresses may have one source and the lateral movement another.

Figs. 66 and 67 are given as examples of the values of the stresses at the two edges of the base of rail, the one on the Reading Company showing relatively small lateral bending and the other on the Richmond, Fredericksburg and Potomac Railroad, relatively large. The values include every readable record in all the runs and all the instruments. On the track of the Reading Company, Instruments 41, 42, 43, and 44 are on the left rail and Instruments 51, 52, 53, and 54 are on the right rail, Instrument 41 being opposite Instrument 51, etc., while on the Richmond, Fredericksburg and Potomac Railroad, these two groups of instruments were placed on the right rail and the left rail, respectively. It is seen that the lateral bending stresses may be large or small. The heavier sections may be expected to have lower lateral bending stresses than the lighter ones, although proportionally they may not differ greatly.

The difference between the stress in the outer edge of the base of rail and the mean stress (average of the stresses in the two edges of the base of rail) may be taken as the stress due to lateral bending of the rail and to be indicative of the amount of lateral bending. If it is positive, the rail is bent outward; if negative, inward. It is found that there is a great variation between individual values of these lateral bending stresses and the average value, and there is need of consideration of the amount and source of such variation.

If the stress at the outside edge of the base of rail is divided by the mean stress, the resulting ratio may be used as an indication of the condition and amount of lateral bending. If this stress ratio is greater than unity, the bending is outward; if less, inward. The range of the values of this ratio will indicate the variations in lateral bending. It should be borne in mind, however, that for the heavier rail a given ratio will mean a relatively smaller lateral bending stress than that in a lighter rail for the same stress ratio.

In Figs. 68 to 75, values of the stress ratios have been plotted for both rails, for each instrument, and for each wheel of the locomotives and cars. Space permits giving diagrams for only a small part of the work, but Figs. 68 to 75 will give some knowledge of the ranges in values and the variations that may be expected at different points on the track and for different runs of the train.

It will be seen that the stress ratio found at a given instrument varies from run to run. Examples of such variations in the range are from 0.40 to 0.70, 0.65 to 0.90, 0.85 to 1.15, 1.00 to 1.45, 1.20 to 1.30, and 0.85 to 1.20. The range in general is 0.15 or so on each side of the average value at the given instrument. Sometimes the range is less than this; occasionally it is more.

The foregoing ranges are for the same wheels of one locomotive or one car. Different wheels give different values of the ratio. Different wheels may give values (averages) at the same instrument varying by as much as 0.20, but usually not more than 0.10. It was found in the tests on the Chicago, Mil-

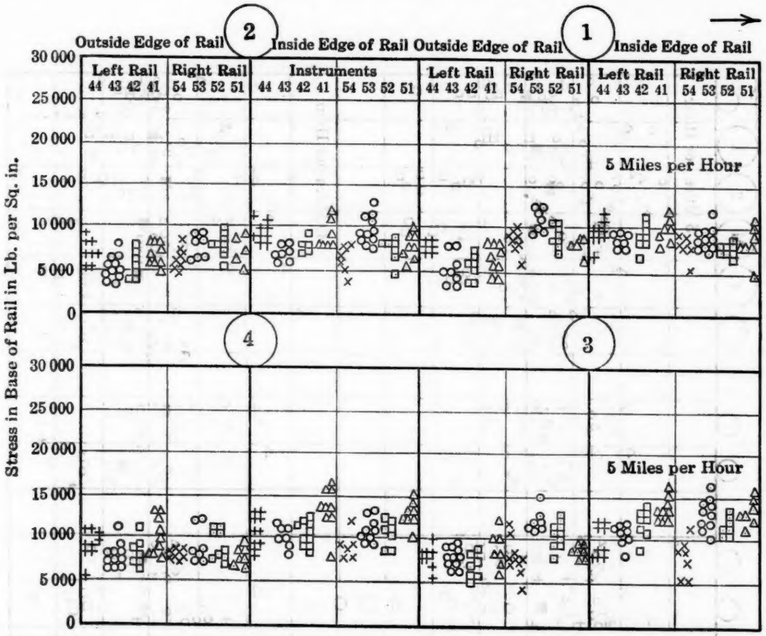


FIG. 66.—OBSERVED VALUES OF STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF RAIL ON STRAIGHT TRACK UNDER THE DRIVERS OF THE MIKADO TYPE LOCOMOTIVE OF THE READING COMPANY. INCLINED TIE-PLATES.

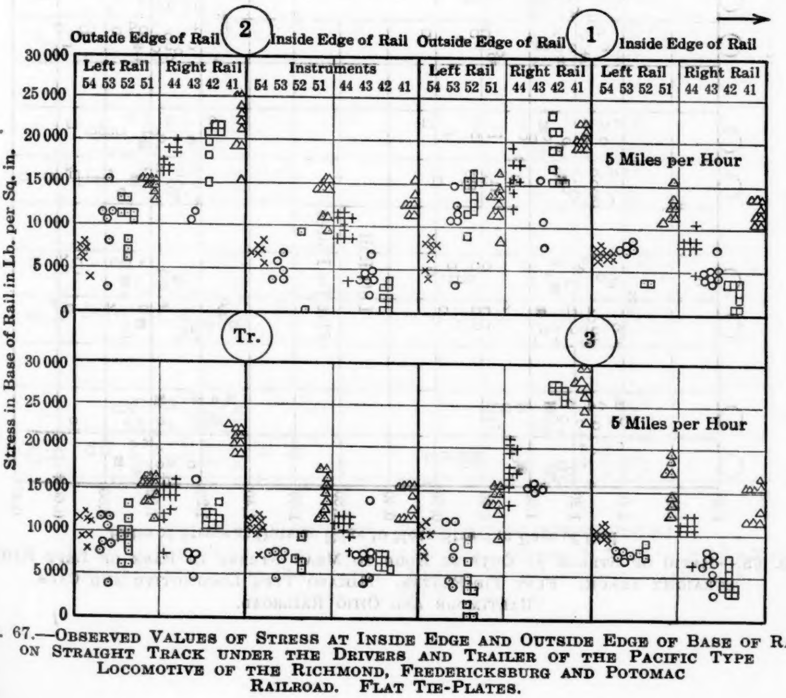


FIG. 67.—OBSERVED VALUES OF STRESS AT INSIDE EDGE AND OUTSIDE EDGE OF BASE OF RAIL ON STRAIGHT TRACK UNDER THE DRIVERS AND TRAILER OF THE PACIFIC TYPE LOCOMOTIVE OF THE RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD. FLAT TIE-PLATES.

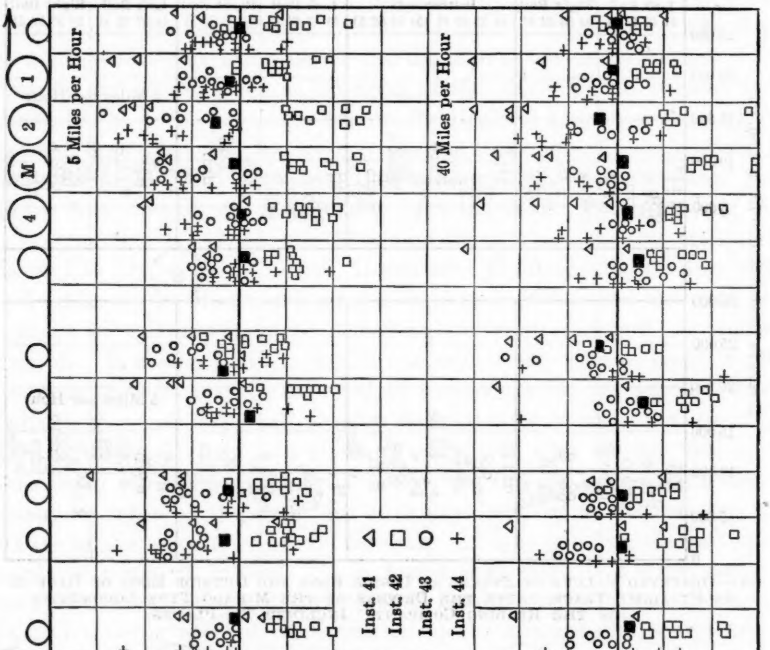


FIG. 68.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF LEFT RAIL ON STRAIGHT TRACK. FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, BALTIMORE AND OHIO RAILROAD.

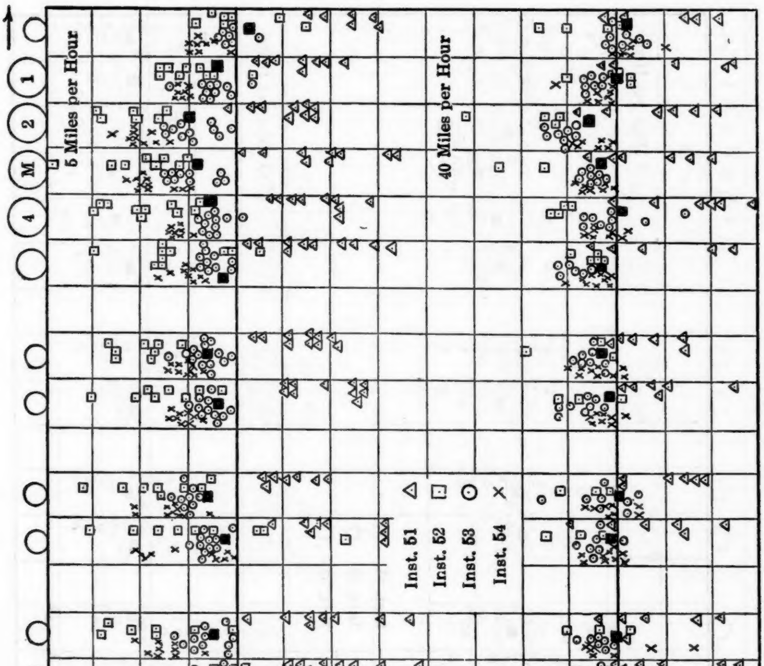


FIG. 69.—RATIO OF STRESS AT OUTERSIDE EDGE TO MEAN STRESS IN BASE OF RIGHT RAIL ON STRAIGHT TRACK. FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, BALTIMORE AND OHIO RAILROAD.

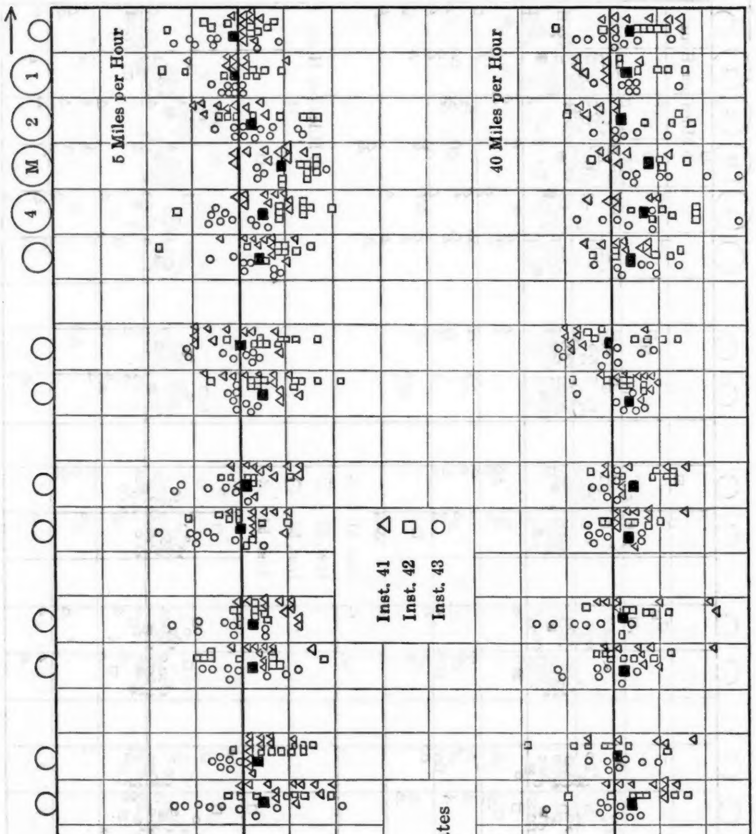


FIG. 70.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF LEFT RAIL ON STRAIGHT TRACK. INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, BALTIMORE AND OHIO RAILROAD.

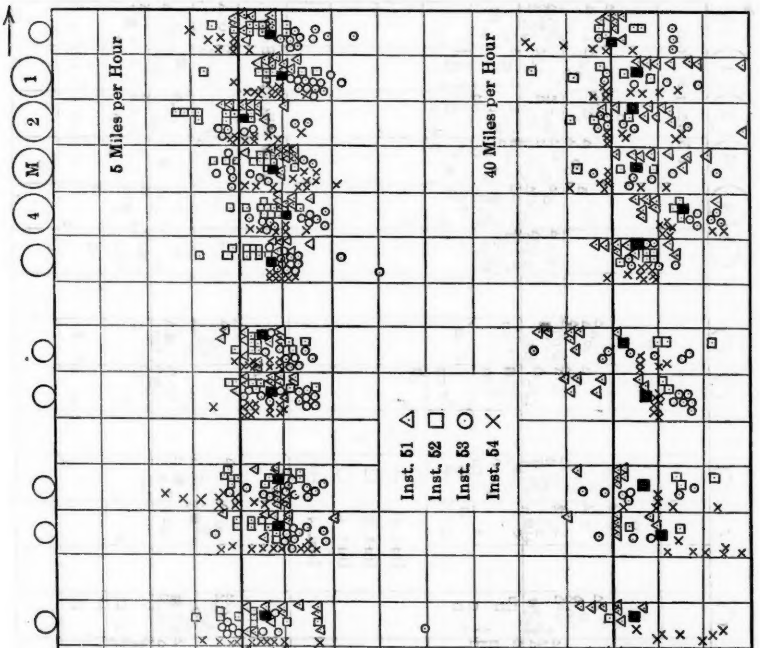


FIG. 71.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF RIGHT RAIL ON STRAIGHT TRACK. INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, BALTIMORE AND OHIO RAILROAD.

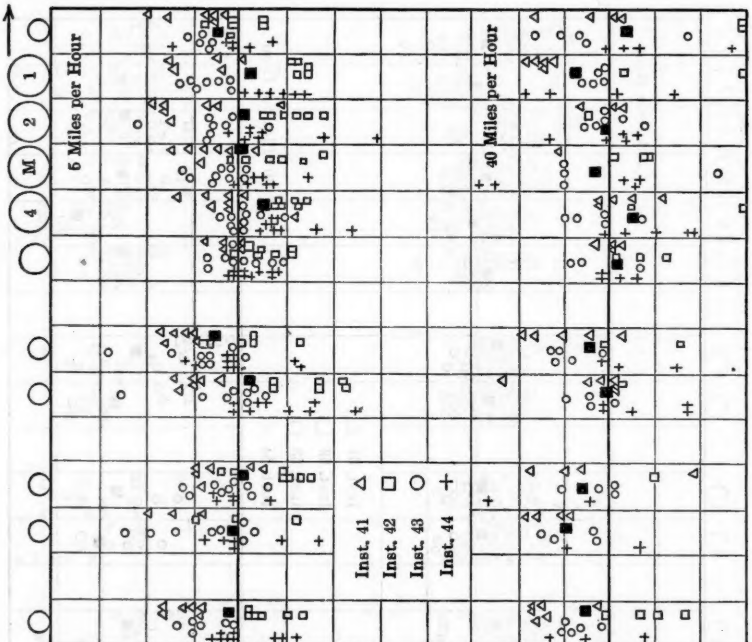


FIG. 72.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF LEFT RAIL ON STRAIGHT TRACK. FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, READING COMPANY.

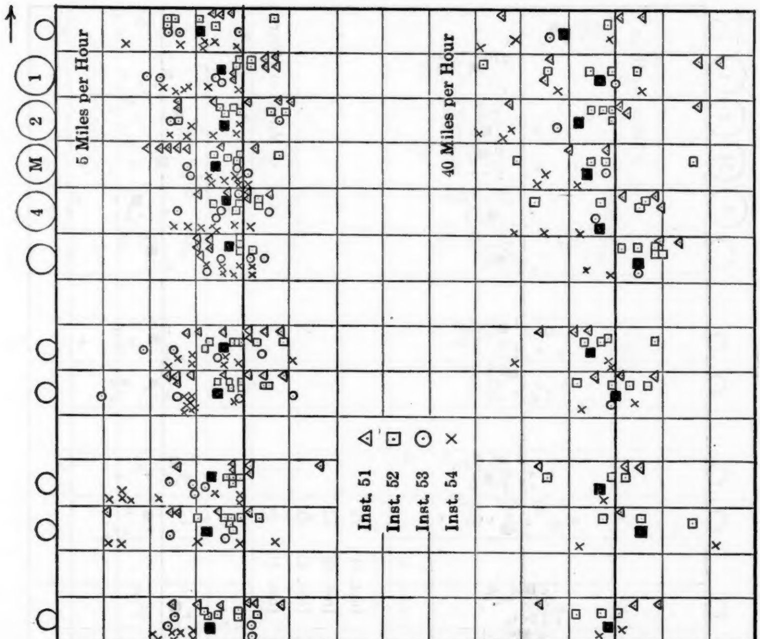


FIG. 73.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF RIGHT RAIL ON STRAIGHT TRACK. FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, READING COMPANY.

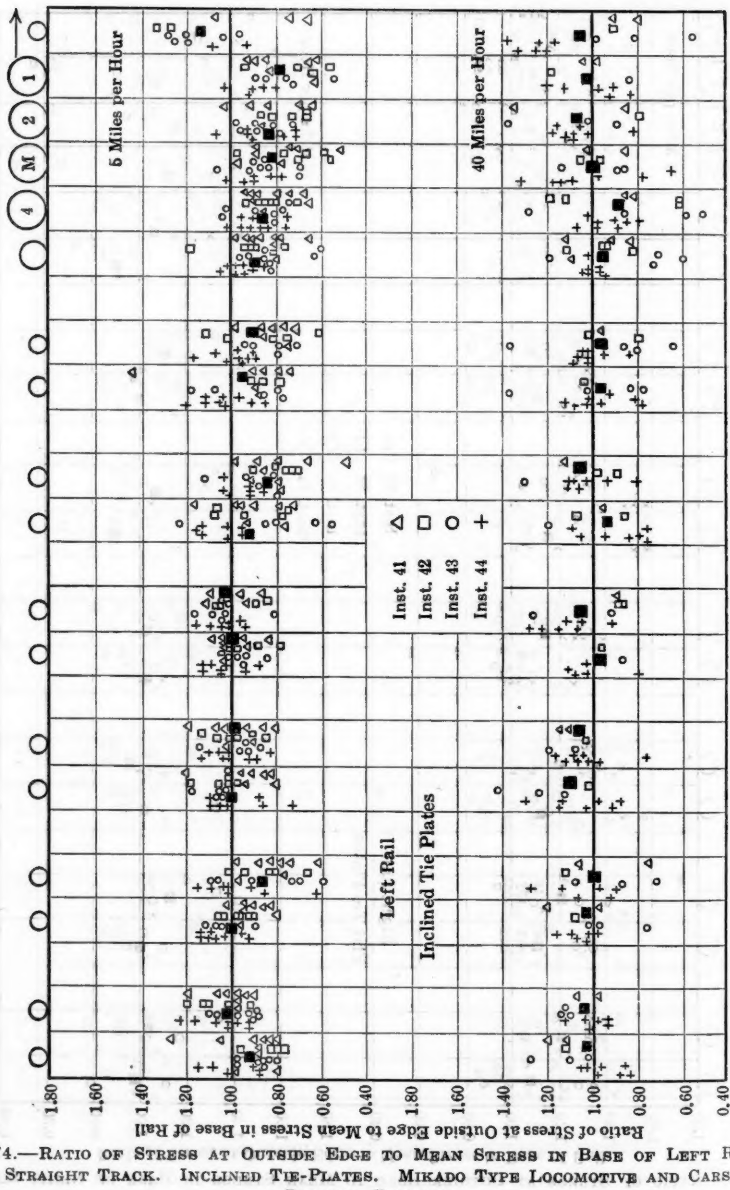


FIG. 74.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF LEFT RAIL ON STRAIGHT TRACK. INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, READING COMPANY.

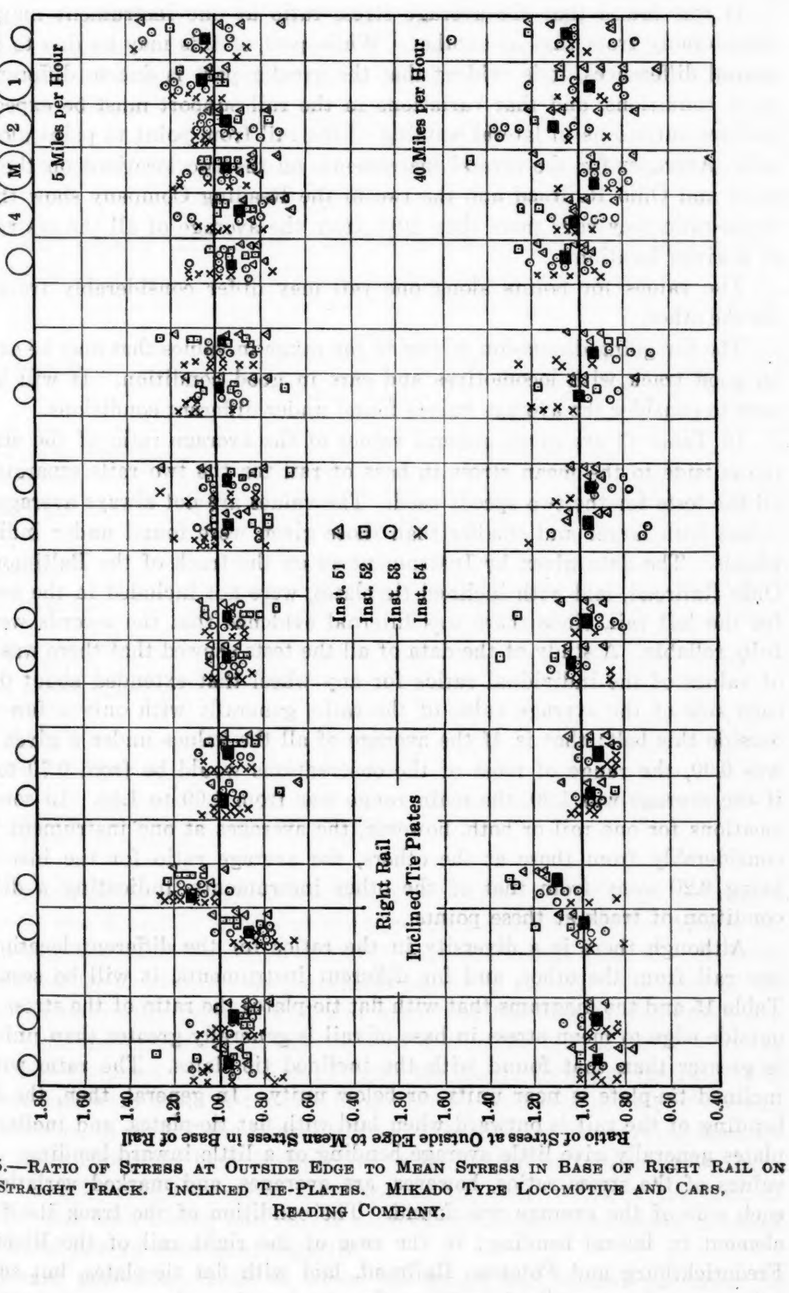


FIG. 75.—RATIO OF STRESS AT OUTSIDE EDGE TO MEAN STRESS IN BASE OF RIGHT RAIL ON STRAIGHT TRACK. INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, READING COMPANY.

waukee and St. Paul Railway that wheels of one locomotive gave quite different ratios at a given instrument than those of another locomotive.

It was found that the average stress ratio at one instrument may differ considerably from that at another. While part of this may be due to instrumental differences, it is evident that the greater part is due to differences in track conditions, and that variations in the rail support must be expected to produce variations in lateral bending of the rail from point to point along the rail. Averages for the several instruments on the two locations on the Baltimore and Ohio Railroad and the two of the Reading Company show that the stress ratio may vary more than 20% from the average of all the instruments at a given location.

The values for points along one rail may differ considerably from those for the other.

The foregoing discussion relates to the range in values that may be expected on good track with locomotives and cars in good condition. It will be best next to consider the average values found under different conditions.

In Table 15 are given general values of the average ratio of the stress at the outside to the mean stress in base of rail for the two rails separately for all the tests for the two speeds used. The values are not always averages, and values both larger and smaller than those given were found under individual wheels. The data given by Instrument 44 on the track of the Baltimore and Ohio Railroad, laid with inclined tie-plates, were not included in the averages for the left rail, since there was internal evidence that the records were not fully reliable. A study of the data of all the tests showed that there was a belt of values of the individual ratios for any wheel that extended about 0.20 on each side of the average value of the ratio, generally with only a few values outside this belt; that is, if the average of all the values under a given wheel was 0.90, the range of most of the observations would be from 0.70 to 1.10; if the average was 1.20, the main range was from 1.00 to 1.40. In some test locations for one rail or both, however, the averages at one instrument varied considerably from those at the others, the average ratio for the instrument being 0.20 away from that of the other instruments, indicating a different condition of track at these points.

Although there is a diversity in the ratios for the different locations for one rail from the other, and for different instruments, it will be seen from Table 15 and the diagrams that with flat tie-plates the ratio of the stress of the outside edge to mean stress in base of rail is generally greater than unity and is greater than that found with the inclined tie-plates. The ratio with the inclined tie-plate is near unity, or below unity. In general, then, the lateral bending of the rail is outward when laid with flat tie-plates, and inclined tie-plates generally give little average bending or a little inward bending. These values of the stress ratios, however, are averages, and marked variations on each side of the average was found. The condition of the track itself is an element in lateral bending; in the case of the right rail of the Richmond, Fredericksburg and Potomac Railroad, laid with flat tie-plates, but actually being canted outwardly 1 in 30 at the test location, the average stress ratio

at all the wheels was in 1.25, the greatest average outward bending found under any of the test trains.

It may be noted that although the average values at the higher speed are somewhat closer to unity than those at 5 miles per hour, the range in ratios is much the same and the variation in results for different runs, different wheels, and different instruments is as great.

TABLE 15.—RATIO OF THE STRESS AT THE OUTSIDE EDGE TO MEAN STRESS IN BASE OF RAIL ON STRAIGHT TRACK.

(A value greater than unity means outward bending; less than unity, inward bending.)

Location.	Speed, in miles per hour.	Left Rail.			Right Rail.		
		Cars.	Tender.	Locomotive.	Cars.	Tender.	Locomotive.
Baltimore and Ohio Railroad:							
Flat tie-plates	5	1.06*	1.03*	1.02*	1.05*	1.07*	1.11*
	40	0.94*	0.98*	0.99*	1.02*	1.03*	1.03*
Inclined tie-plates	5	0.96†	0.97†	0.94†	0.88	0.88	0.87
	40	0.95†	0.95†	0.90†	0.96	0.86	0.88
Reading Co.:							
Flat tie-plates	5	1.05	1.01	0.98	1.14	1.18	1.11
	40	1.05	1.09	1.00	1.02	1.02	1.09
Inclined tie-plates	5	0.98	0.91	0.89	0.97	0.97	0.97
	40	1.04	0.99	1.00	0.99	0.96	1.02
Lehigh Valley, Railroad:							
Inclined tie-plates	5	0.99	1.00	0.93	0.92	0.97	0.97
Richmond, Fredericksburg and Potomac Railroad.:							
Flat tie-plates	5	1.10	1.14	1.01	1.29	1.27	1.32
	40	1.09	1.08	1.09	1.24	1.33	1.35
Inclined tie-plates	5	1.04	1.02	1.04	1.03	1.13	1.11
	40	0.97	0.93	1.00	1.07	1.08	1.18

* Includes one instrument much below average.
† Instrument 44 omitted.

24.—The Tilting and Lateral Movement of the Rail Under Load.—Straight Track.—It was found by means of the apparatus described under Article 21, "Conduct of Tests and Reduction of Data", that on straight track the rail itself tilted outwardly or inwardly as the wheels passed by and that the head of the rail moved outwardly or inwardly, and even that the base of the rail had a slight lateral movement in addition to that due to the lateral bending of the rail already discussed. Figs. 76 and 77 give the lateral tilting of the rail. The measurements were made at the side of the head ½ in. below the top of the rail and represent, in inches, the lateral tilting of the rail for the vertical distance of this point above the base, irrespective of the lateral movement of the base. Figs. 78 and 79 give the lateral movement of the same point on the head of the rail. For both tilting and lateral movement, outward movement is called positive and inward movement negative.

Outward movement tends to widen the gauge and inward movement to narrow it. The general movement of the base of rail may be found by sub-

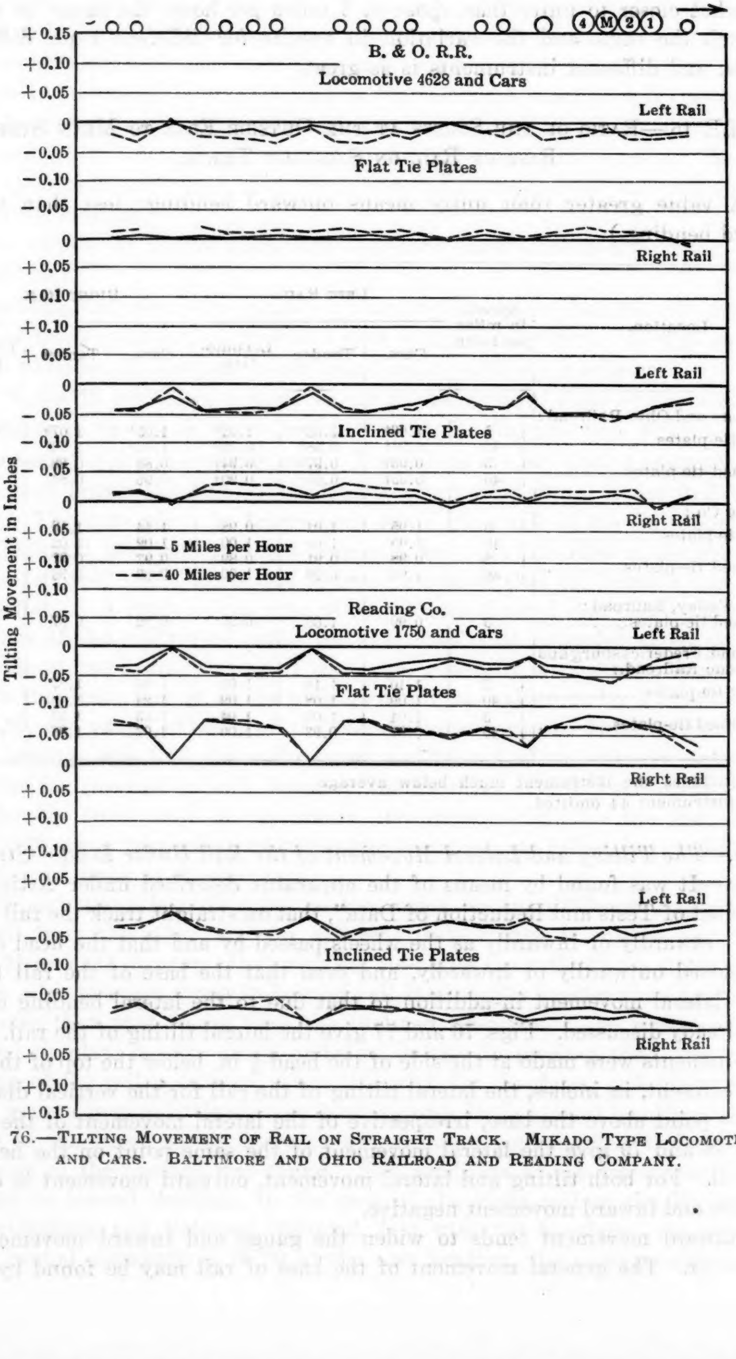


FIG. 76.—TILTING MOVEMENT OF RAIL ON STRAIGHT TRACK. MIKADO TYPE LOCOMOTIVES AND CARS. BALTIMORE AND OHIO RAILROAD AND READING COMPANY.

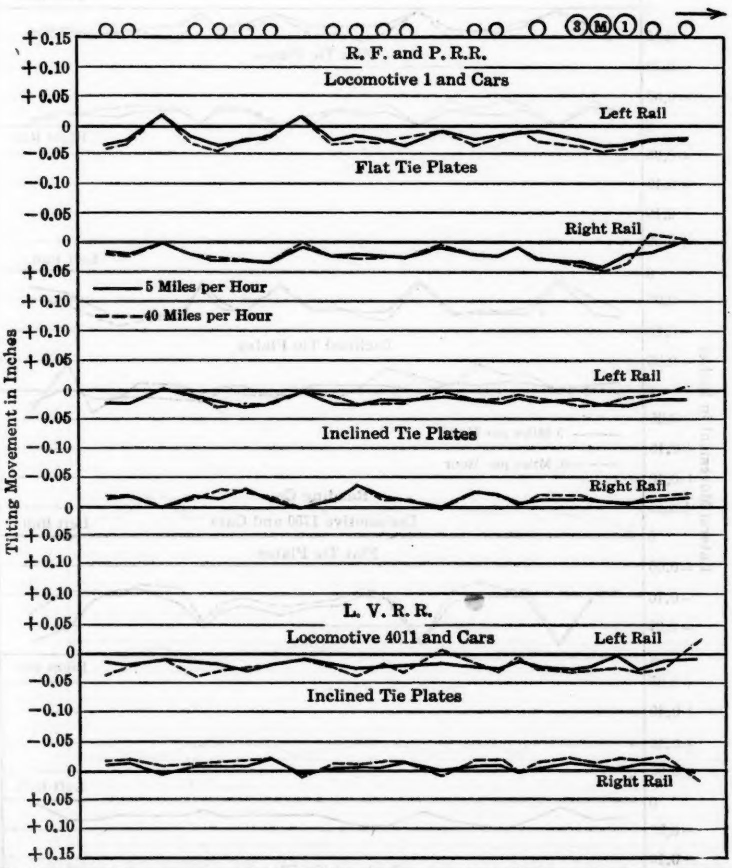


FIG. 77.—TILTING MOVEMENT OF RAIL ON STRAIGHT TRACK. PACIFIC TYPE LOCOMOTIVE AND CARS. RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD. SANTA FÉ TYPE LOCOMOTIVE AND CARS, LEHIGH VALLEY RAILROAD.

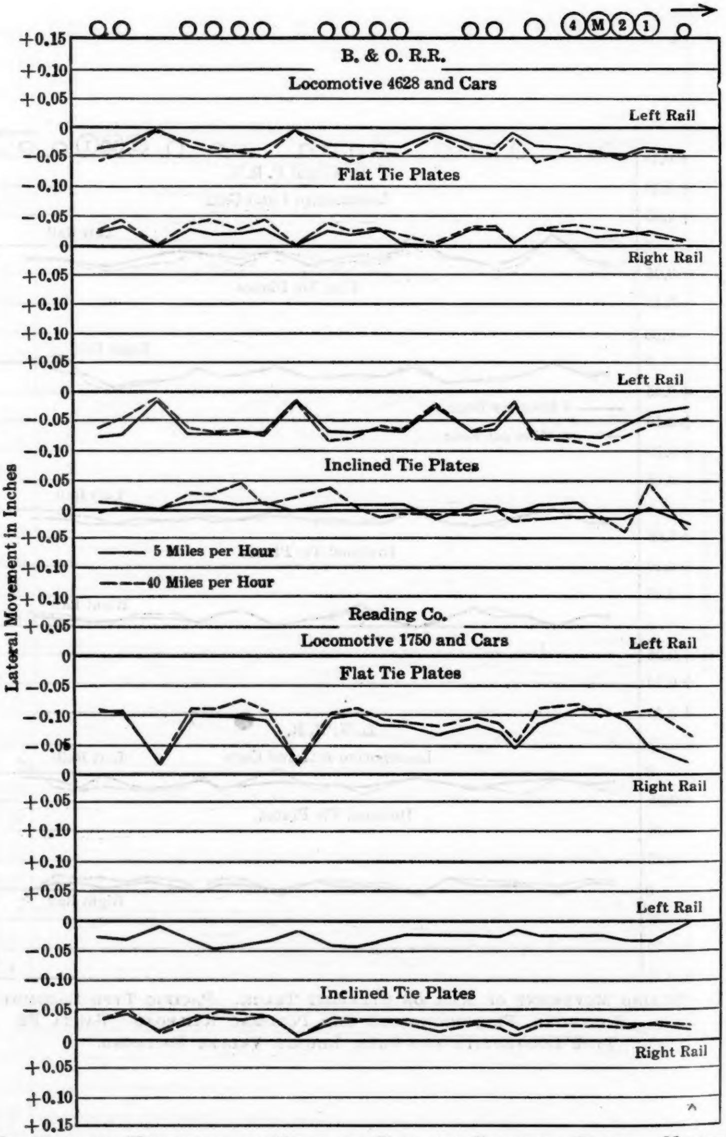


FIG. 78.—LATERAL MOVEMENT OF HEAD OF RAIL ON STRAIGHT TRACK. MIKADO TYPE LOCOMOTIVES AND CARS. BALTIMORE AND OHIO RAILROAD AND READING COMPANY.

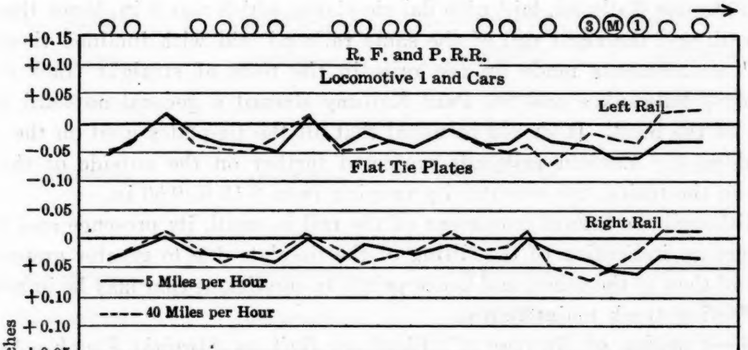


Fig. 79.—Lateral Movement of Head of Rail on Straight Track. Pacific Type Locomotive and Cars. Richmond, Fredericksburg and Potomac Railroad. Santa Fé Type Locomotive and Cars, Lehigh Valley Railroad.

tracting algebraically the amount of the tilting movement from that of the movement of the head of the rail. For both movements, the values given are averages of two to four runs.

It will be seen that the lateral movement of the head is generally greater than that of the tilting action, the former ranging from, say, 0.04 to 0.10 in., and the latter from 0.02 to 0.06 in. The movement of the rail was inward in all cases except in that of the right rail of the Richmond, Fredericksburg and Potomac Railroad, laid with flat tie-plates, which was 2 in. lower than the left rail, and the right rail of the same railroad laid with inclined tie-plates. The measurements made on the rails in the tests of straight track of the Chicago, Milwaukee and St. Paul Railway showed a general outward movement of the head. It should be noted that all the tie-plates used on the track tested on the Eastern railroads projected farther on the outside of the rail than on the inside, the eccentricity ranging from 0.15 to 0.50 in.

Although the lateral movement of the rail is small, its presence and direction are an indication of the tilting of the tie-plate due to greater pressure at one end than at the other, and hence points to conditions that may be important as affecting track maintenance.

25.—Position of Bearing of Wheel on Rail on Straight Track.—As has already been described, a copper wire laid across the rail was pulled a short distance between the passage of one wheel and the next and again at the next interval, the wire flattened by the wheel giving information on the position of the bearing of the wheel on the rail, the impression left on the wire by punch-marks made on the rail in advance permitting identification of the position of the wire on the rail. In Figs. 80 to 84 are plotted results of tests with the copper wire. The width of the flattened portion at various points along the length of the impression was measured and the initial diameter of the wire was subtracted, an arbitrary procedure; the remainder was plotted above the rail in the diagrams. The plot so made does not measure the intensities of pressure of wheel on rail, but the position of the greatest ordinate or perhaps the center of gravity of the area may be considered to be the point of greatest pressure, and the bearing contact may be expected to extend a short distance on either side of this point, but not to the limit of the area so plotted. The intensity of pressure will be greatest at the point of greatest ordinate and decrease rapidly each way, the width of the bearing being much less than represented in the diagram. It should also be noted that an impression shown at the gauge side of the head may mean merely that the flange was close to the rail; only a considerable width of the area is indicative of flange pressure.

For the test location of the Baltimore and Ohio Railroad, with inclined tie-plates (cant of rail 1 in 15 and 1 in 17), the center of pressure between wheel and rail for both locomotive and cars was at or near the middle of the the head of the rail. The brightness was spread well over the full width of the head of the rail. With the flat tie-plates (rails canted inwardly about 1 in 60), the center of bearing for both locomotive and cars was on an average about 0.6 in. away from the middle of the head and toward the gauge side.

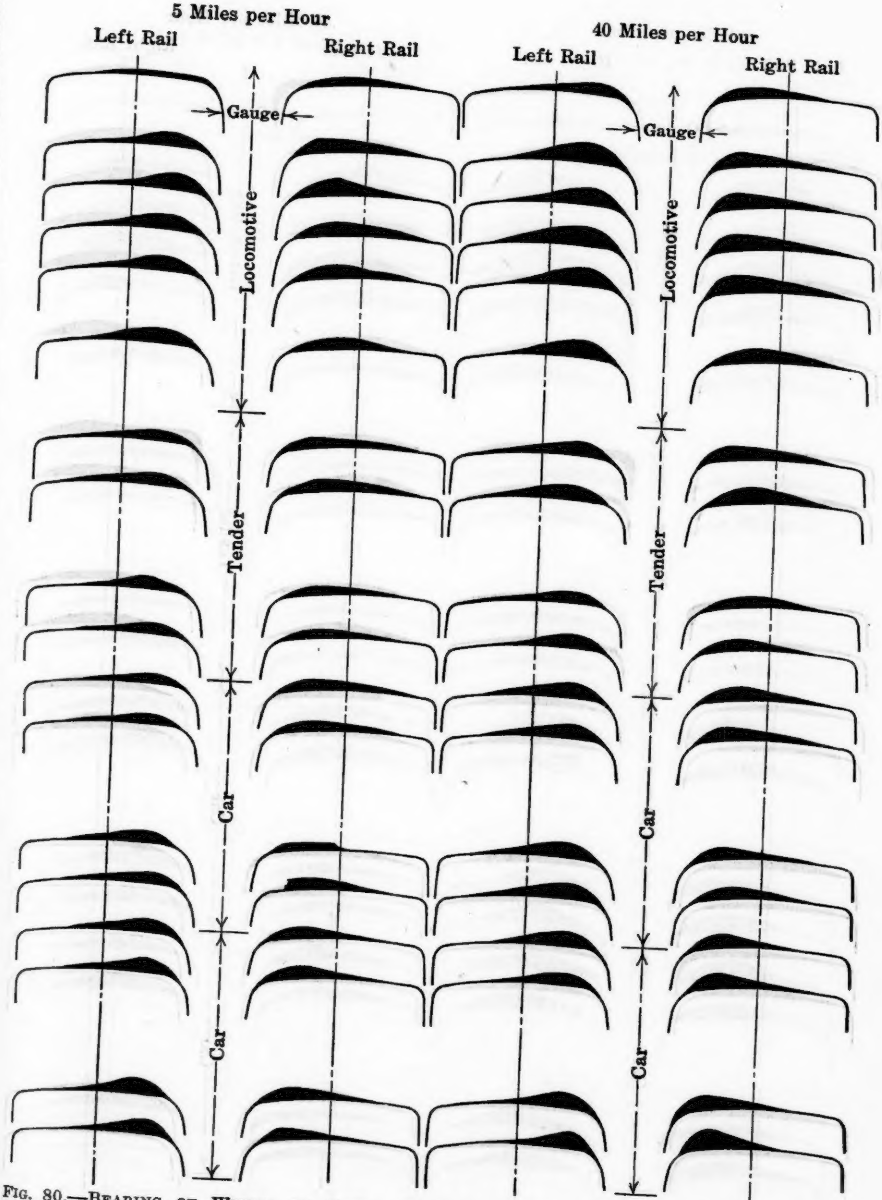


FIG. 80.—BEARING OF WHEEL ON RAIL. STRAIGHT TRACK. FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS. BALTIMORE AND OHIO RAILROAD.

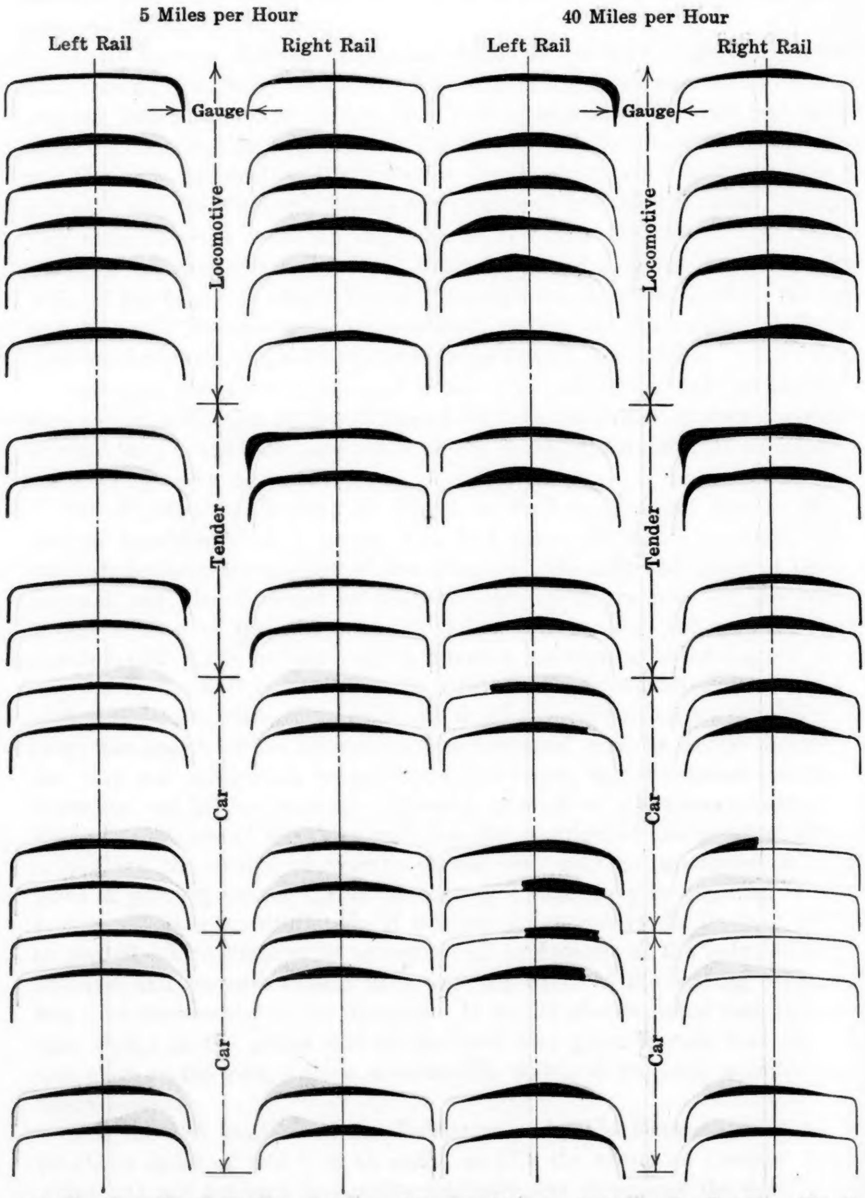


FIG. 81.—BEARING OF WHEEL ON RAIL. STRAIGHT TRACK. INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS. BALTIMORE AND OHIO RAILROAD.

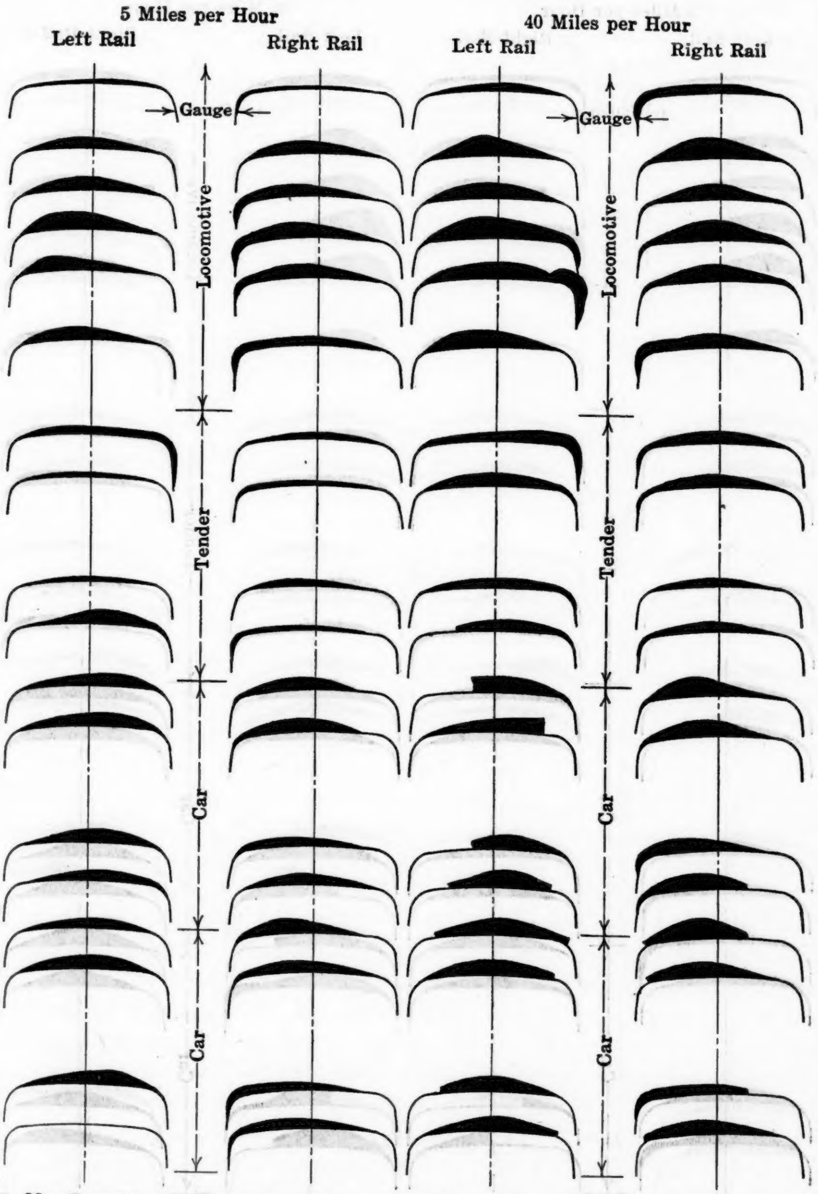


FIG. 82.—BEARING OF WHEEL ON RAIL. STRAIGHT TRACK. FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS. READING COMPANY.

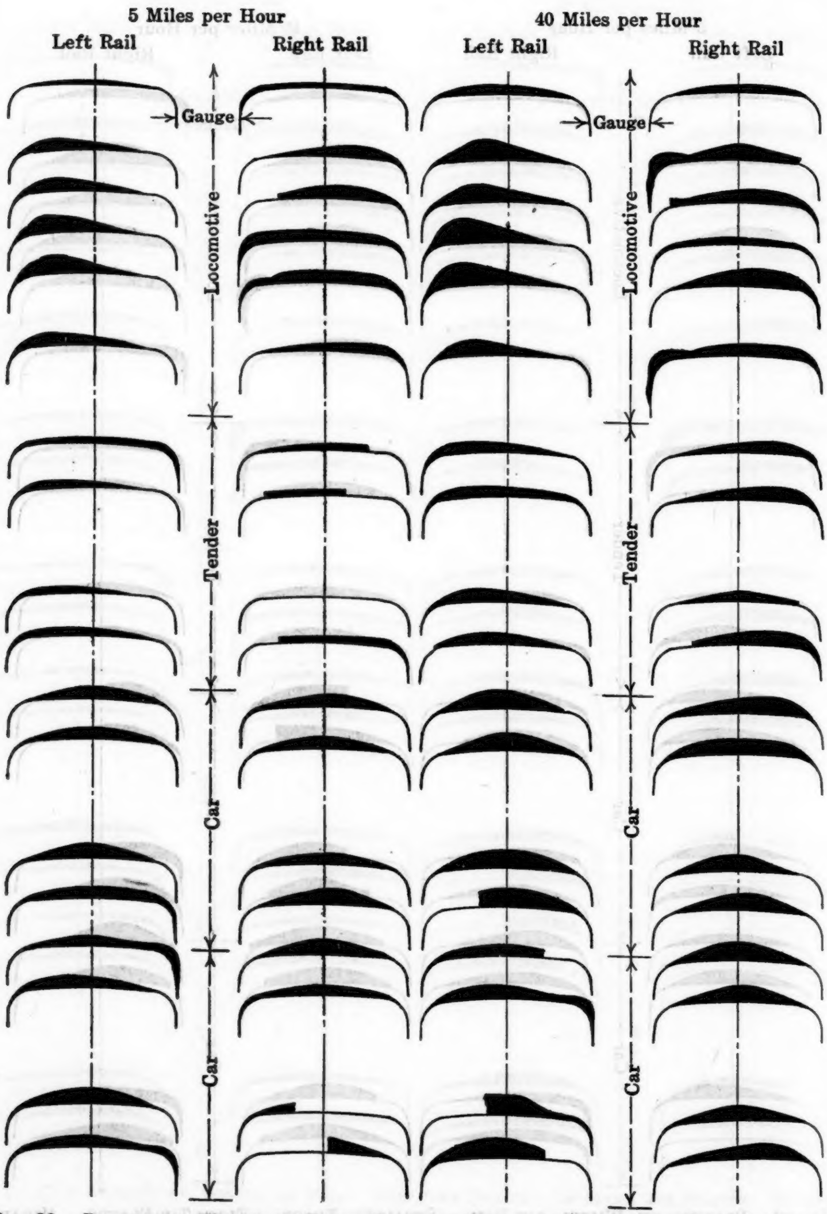


FIG. 83.—BEARING OF WHEEL ON RAIL. STRAIGHT TRACK. INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS. READING COMPANY.

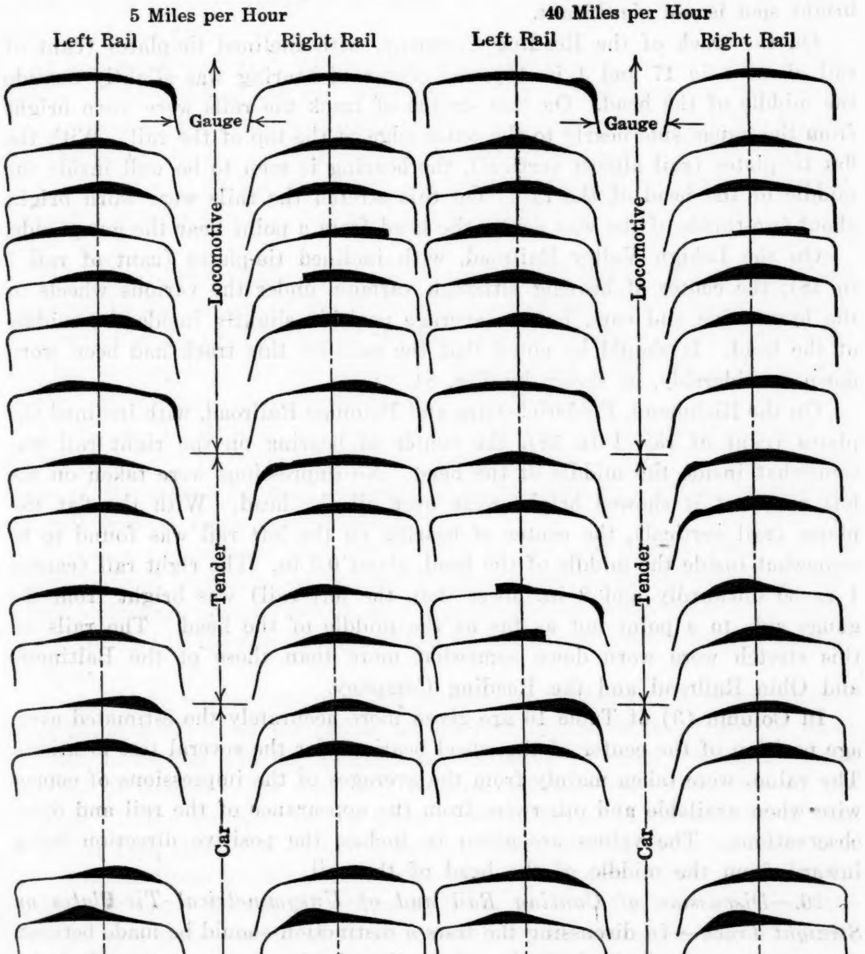


FIG. 84.—BEARING OF WHEEL ON RAIL. STRAIGHT TRACK. INCLINED TIE-PLATES: SANTA FÉ TYPE LOCOMOTIVE AND CARS. LEHIGH VALLEY RAILROAD.

As this rail had been on flat tie-plates only a few days, the position of the bright spot is not significant.

On the track of the Reading Company, with inclined tie-plates (cant of rail about 1 in 17 and 1 in 16), the center of bearing was slightly outside the middle of the head. On this stretch of track the rails were worn bright from the gauge side nearly to the outer edge of the top of the rail. With the flat tie-plates (rail almost vertical), the bearing is seen to be well inside the middle of the head of the rail. On this stretch the rails were worn bright about two-thirds of the way across the head from a point near the gauge side.

On the Lehigh Valley Railroad, with inclined tie-plates (cant of rail 1 in 18), the center of bearing although variable under the various wheels of the locomotive and cars, has an average position slightly inside the middle of the head. It should be noted that the rails on this track had been worn down considerably, as shown by Fig. 84.

On the Richmond, Fredericksburg and Potomac Railroad, with inclined tie-plates (cant of rail 1 in 18), the center of bearing on the right rail was somewhat inside the middle of the head. No impressions were taken on the left rail, but it showed bright wear over all the head. With the flat tie-plates (rail vertical), the center of bearing on the left rail was found to be somewhat inside the middle of the head, about 0.3 in. The right rail (canted 1 in 30 outwardly and 2 in. lower than the left rail) was bright from the gauge side to a point not as far as the middle of the head. The rails on this stretch were worn down somewhat more than those of the Baltimore and Ohio Railroad and the Reading Company.

In Column (3) of Table 16 are given more accurately the estimated average position of the center of the wheel bearings for the several test locations. The values were taken mainly from the averages of the impressions of copper wire when available and otherwise from the appearance of the rail and other observations. The values are given in inches, the positive direction being inward from the middle of the head of the rail.

26.—Discussion of Canting Rail and of Unsymmetrical Tie-Plates on Straight Track.—In discussing the tests a distinction should be made between the purpose and effect of canting the rail and of using unsymmetrical tie-plates. It will be evident that the two give separate and apparently not closely related effects.

A study of the results shows that for the rails laid on inclined tie-plates the averages of the ratios of stress in outer edge to mean stress in base of rail are either close to unity or less than unity, and that the averages of the ratios for the rails laid with flat tie-plates is greater than unity and greater than those found in rails laid with the inclined tie-plates. In other words, the lateral bending of the rail canted inwardly of the track on the average is very small or the bending is inward, the latter being found when the rail is canted more than 1 in 20; and for rail laid vertically the outward lateral bending stresses on the average are quite marked. It may be expected that the maintenance of track affected by lateral bending of rail will be

influenced by the two classes of tie-plates in much the same way as the lateral bending of the rail.

It is also evident that the bearing of wheels on the rail is closer to the middle of the head of the rail when the rail is canted inwardly than when it is vertical. This result was quite marked and was common for the drivers and the wheels of the tenders and the loaded freight cars. The distribution of brightness and of the wear over the head of the rail on the track tested covered a greater width of the head for the canted rail than for the vertical rail.

The actual cant of the rail as measured in the track differed from that indicated by the design of the tie-plates, whether they were flat or inclined. This variation in the position of the rail may be expected to be attributable to other sources than the inclination of the tie-plate itself—probably, as will be seen, to the amount of eccentricity of the tie-plate.

As far as the data of the tests are conclusive, an inward inclination of the tie-plate of 1 in 20 is effective on straight track in reducing the average value of the lateral bending stresses in the rail and in securing bearing on the rail that is fairly near central.

There is no evidence that the canting of the rail has an effect on the general lateral movement of the rail or on changes in gauge, either narrowing or widening, except possibly as a slightly greater eccentricity of tie-plate may be needed in the case of the inclined tie-plate to prevent outward tilting of the rail.

It should be remembered that the average lateral bending stresses referred to as being small when the rail is canted 1 in 20 and as being greater when the rail is vertical are the averages of a number of runs. Whether or not the rail is canted there will still be the variation on either side of the average. If the average ratio of stress at outside edge to mean stress in base of rail is 1.00, there will still be a belt of values ranging from as much as a 20% greater stress at the outside edge to a 20% greater stress at the inside edge, with many occasional values as great as 30% more or less than the average, and some even greater variations. If the average ratio with a flat tie-plate were 1.10, the corresponding upper range would reach to a stress at the outside edge of base of rail 30% greater than the mean stress for the ordinary upper limit, with other still greater additional stresses. Even if the average lateral bending stress is small, it is seen that there will still be need for adequate lateral strength and stiffness in the rail. The use of tie-plates with the proper inclination will not remedy this situation.

Independently of the lateral bending of the rail just considered, as the load passes by there is a tilting of the rail inwardly or outwardly due to the greater bearing pressure on the tie under the tie-plate at one side of the rail or the other, and the consequent greater depression of the tie-plate at that end. If the load were applied to the middle of the head and were vertical, the design of the tie-plate should be symmetrical with respect to the rail. When the resultant force applied to the rail is inclined outwardly of the track, the use of a symmetrical tie-plate may result in an outward tilting of the rail

and a widening of the gauge. The use of an unsymmetrical tie-plate having the greater projection on the outside of the rail will tend to correct this condition. Too great an eccentricity in the tie-plate will result in an inward tilting of the rail. The modulus of elasticity of the wood of the tie when compressed across the grain is small in comparison with that lengthwise of the grain, say, one-fiftieth part, if the detrusion of the tie-plate due to bearing pressure is included. As a result, small differences in the distribution of pressure along the tie-plate will throw it out of level and result in a tilting of the rail and, finally, in a permanent depression of one end of the tie-plate and a cutting of the tie. What eccentricity of tie-plate will be most satisfactory is a matter to be determined by experience. The observations made on the tilting of the rail in the tests, however, may be of service in judging of what may be considered a proper value.

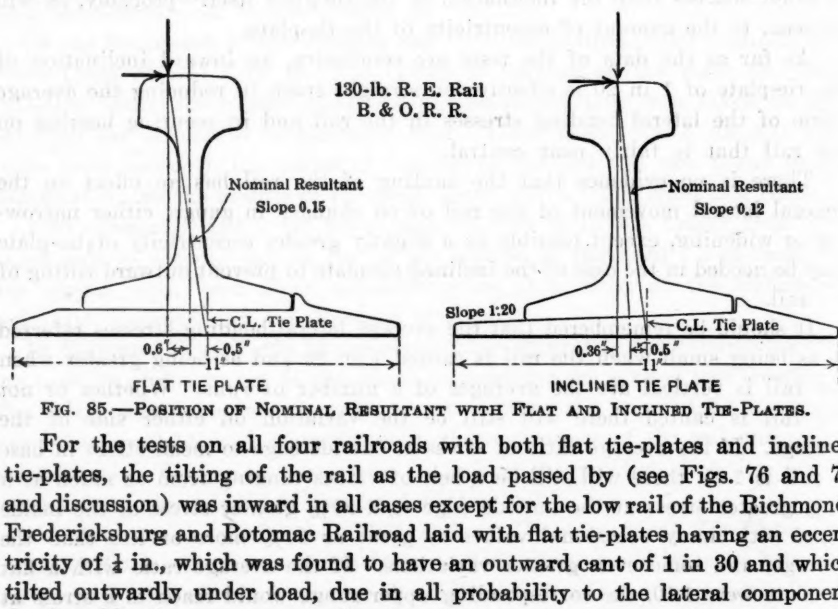


Fig. 85.—Position of Nominal Resultant with Flat and Inclined Tie-Plates.

For the tests on all four railroads with both flat tie-plates and inclined tie-plates, the tilting of the rail as the load passed by (see Figs. 76 and 77 and discussion) was inward in all cases except for the low rail of the Richmond, Fredericksburg and Potomac Railroad laid with flat tie-plates having an eccentricity of ½ in., which was found to have an outward cant of 1 in 30 and which tilted outwardly under load, due in all probability to the lateral component developed by the transverse inclination of the track. Almost without exception on all the track the inward inclination of the rail from the vertical was greater than that due to the form of the tie-plate. It would seem then that the increase in the cant of the rail is due to continued tilting under the passage of trains and uneven pressure on the tie. The tests made heretofore on track with symmetrical tie-plates and without tie-plates have shown a slight outward movement of the rail, and it is common experience that the gauge on such track generally widens.

It will be well also to study the position of the center of the tie-plate with respect to that of the bearing of the wheel on the rail. To bring out the position of the forces acting, Fig. 85 has been prepared, which shows the section of the 130-lb. rail of the Baltimore and Ohio Railroad with the flat tie-plate and the inclined tie-plate.

The eccentricity of the tie-plate in both cases (distance from the center of the rail to the center of the plate) is 0.5 in. The inclination of the inclined tie-plate is 1 in 20. If the vertical force is applied to the head of the rail at a point 0.6 in. inside the middle of the head in the case of the vertical rail and at the middle of the head of the canted rail, and if the amount of the lateral force in each case is such that the resultant passes through the center of the tie-plate, the real eccentricity of the force for the two cases would be 1.10 and 0.86 in., respectively, and the slope of the resultant (tangent of the angle with the vertical) 0.15 and 0.12, respectively, or 1 horizontal to 6.8 and 8.4 vertical. Such slopes are greater than would be expected from the values of the average lateral bending stresses in any of the tests found thus far. It would appear then that the resultant would fall inside the center of the tie-plate in such cases.

TABLE 16.—CENTER OF WHEEL-BEARING, ECCENTRICITY OF TIE-PLATE, CANT OF RAIL, NOMINAL ECCENTRICITY, AND SLOPE OF NOMINAL RESULTANT ON STRAIGHT TRACK.

(The values are given in inches, the positive direction being inward from the middle of the head of rail. The position of the center of the wheel bearing was estimated from the impressions of copper wire for the wheels of locomotives and cars when available and otherwise from the appearance of the rail and other observations. The nominal eccentricity is the horizontal distance from the middle of the length of the tie-plate to a vertical line drawn through the center of the wheel bearing. (See Fig. 85.) The so-called nominal resultant is assumed to be acting along a line joining the center of the wheel bearing and the middle of the tie-plate, and the slope given is that between this line and the vertical (tangent of the angle)).

Location. (1)	Average cant of rail. (2)	Center of wheel bearing. (3)	Eccentricity of tie-plate. (4)	Eccentricity due to cant of rail. (5)	Nominal eccentricity. (6)	Slope of nominal resultant. (7)
Baltimore & Ohio Railroad:						
Flat tie-plate	1:60	0.60	0.50	0.12	1.22	0.16
Inclined tie-plate.........	1:16	0.00	0.50	0.45	0.95	0.13
Reading Co.:						
Flat tie-plate	1:200	0.30	0.50	0.04	0.84	0.11
Inclined tie-plate.........	1:16	—0.10	0.50	0.45	0.85	0.12
Lehigh Valley Railroad:						
Inclined tie-plate*........	1:18	0.00	0.15	0.42	0.57	0.08
Richmond, Fredericksburg & Potomac Railroad:						
Flat tie-plate	00	0.30	0.25	0.00	0.55	0.09
	—1:30	0.70	0.50	—0.21	0.74	0.12
Inclined tie-plate.........	1:20	0.10	0.50	0.33	0.93	0.14

* Worn rail.

Table 16 gives information on the test track from which the possible eccentricity of the resultant force may be studied. Fig. 85 will help in understanding the terms used. The center of the wheel-bearing with respect

to the middle of the head of the rail, as judged from the impressions of copper wire and the wear and brightness of the rail, the eccentricity of the tie-plate (distance from the center of base of rail to the mid-point of the tie-plate), and the eccentricity due to cant of rail are given. The position of the resultant of the gravity load and the lateral force on the rail will be unknown. A force passing through the center of the wheel bearing and the mid-point of the tie-plate will be called the nominal resultant. This nominal resultant will give equilibrium only in case the average lateral force acting upon the rail is of such magnitude as to make the resultant pass through the mid-point of the tie-plate. If the nominal resultant were the actual resultant, the pressure on the tie-plate would be uniformly distributed over it. The slope of this nominal resultant and the corresponding nominal eccentricity of such a line are given in Table 16. A study of the eccentricities and the tilting of the rail under load may throw light on the position of the actual resultant.

In comparing the nominal eccentricity given in Table 16, it should be remembered that the flat tie-plate on the Baltimore and Ohio Railroad had only recently been laid and, therefore, that no change in cant had occurred with these tie-plates; no direct conclusion can be drawn concerning the nominal eccentricity, 1.22 in., except as may be judged from the tilt of the rail under load. The rail on the test track on the Lehigh Valley Railroad was considerably worn and, therefore, the increase in cant in that rail may not have been caused by conditions existing at the time of the test. As the track of the Richmond, Fredericksburg and Potomac Railroad, laid with flat tie-plates, has one rail 2 in. lower than the other, the cant of the two rails must have been influenced by the transverse inclination of the track. However, it is seen that in nearly every case the cant of the rail inward had been increased by the traffic from that at which presumably it was laid.

From the data available it may be concluded that for both flat tie-plates and inclined tie-plates the tie-plates should be unsymmetrical. It may be concluded also that with the rails and tie-plates used, an eccentricity of 0.5 in. is too great. Possibly ¼ in. may be nearer the proper amount. It would seem better to state the eccentricity of the combination in terms of the position of the wheel bearing, the cant of the rail, and the eccentricity of the tie-plate itself. For the size of rail and tie-plate used in the track tested, it may be judged that 0.75 in. is a proper value for the nominal eccentricity. With the 130-lb. rail this is about one-tenth the height of the rail and tie-plate. With a cant of 1 in 20 and a central wheel bearing on the rail, an eccentricity of tie-plate of ¼ to ⅜ in. would result. Whether this eccentricity of tie-plate will give proper conditions may best be told by experience.

The position of the resultant gravity load and the lateral force referred to has been assumed to be that for the average of a number of applications of load. Due to variations in the lateral pressure on the rail, it will be seen that individual runs will develop resultants intersecting the bottom of the tie-plate at one side or the other of the average position. Such positions of the resultant will give uneven distributions of bearing pressure between tie-plate and tie, a higher intensity of pressure at one edge, and a lower pressure

at the other edge. The increase in intensity will depend not only on the position of the resultant, but also on the length of the tie-plate. As an illustration of change in intensity, consider that the main belt of ratios of stress at edge to mean stress in base of rail (discussed in Article 23, "Lateral Bending Stresses in Straight Track and Ratios of Stress at Outside Edge to Mean Stress in Base of Rail") extends 0.30 on each side of the average ratio. Assuming that the lateral force producing the lateral bending bears the same relation to the vertical load as the lateral bending moment bears to the vertical bending moment developed at the point (an assumption that seems fairly reasonable even though no conclusive information is available), and considering that the section modulus, $\frac{I}{c}$, of the rail about the vertical axis is one-fifth that about the horizontal axis, it is found that the addition of 0.30 to the stress ratio means an addition to the lateral force on the rail equal to 0.06 of the vertical force. This means for the rail and tie-plate shown in Fig. 85 that the resultant will intersect the bottom of the tie-plate 0.06 times $7\frac{1}{2}$ in., a distance of, say, 0.45 in. from the average position. Assuming that the average position of the resultant passes through the middle of the length of the tie-plate and that the tie-plate is 11 in. long, the relative eccentricity of the plate is $\frac{0.45}{11}$. Using Equation (34), it is seen that the intensity of stress at the edge is about 25% greater or less than the average pressure on the tie. A longer tie-plate will give a smaller variation from the average and a shorter tie-plate a greater variation. Whether such a variation is or is not excessive will depend, of course, on whether the average bearing pressure is sufficiently small for the kind of wood in the tie.

Both the matter of low average pressure of plate on tie and that of a sufficiently small variation in intensity of pressure for the more extreme variations in lateral forces against the rail bring emphasis to the need of adequate length of tie-plates. It is apparent that the position of the resultant at the tie is a function of the height of the rail and thickness of the tie-plate, and granted a sufficiently large bearing area, the desirable length of tie-plate may be considered a function of this total height. The dimension, 11 in., is about one and one-half times the height of the rail and the thickness of the tie-plate. Whether this tie-plate is sufficiently long for the loads carried is a matter for experience to determine, but it is evident that long tie-plates and large bearing areas have important advantages.

To recapitulate the discussion of the tests on straight track: The rails laid with an inward cant of 1 in 20 or thereabouts in general show a smaller average lateral bending than those laid in a vertical position. The canted rail on all the railroads shows a bearing between wheel and rail for both locomotive and cars that is close to the middle of the head of the rail, while with the vertical rail the bearing is well on the inside, about 0.5 in. from the middle of the head. The distribution of brightness and of wear extends over a greater width of the head for the canted rail than for the vertical rail. The results indicate an advantage in canting the rail by giving a more

nearly central bearing and a better distribution of wear on the rail and in reducing the average lateral bending stresses in the rail. The indications from the tests are that an inward cant of 1 in 20 is effective and better than a smaller one.

In this connection, it should be noted that the standard coning of the wheels of the locomotive and cars is 1 in 20, except for the Baltimore and Ohio Railroad, for which 1 in 13 is standard. In considering the results on the Baltimore and Ohio Railroad, it should be borne in mind that the average cant of the rails on the test sections of the straight track averaged 1 in 16.

Although the inward canting of the rail reduces the average of the lateral bending stresses, it has no apparent effect on the variation in the lateral bending stresses. There will still be lateral bending stresses ranging to 20 and 30% of the vertical bending stress, with occasional values still greater, the stress in an edge of the base of the rail being increased to this extent over the mean stress in base of rail. The need for adequate lateral strength and stiffness in the rail still remains.

The position of the wheel bearing on the vertical rail, and that on the canted rail, together with the cant of the rail, indicates that in both cases the resultant of the vertical and lateral forces on the average slope away from the vertical. The general inward tilting of the rail in the tests is evidence that for both the flat tie-plates and the inclined tie-plates used the eccentricity of the tie-plates (generally 0.5 in.) is too great, making, as it does, with the eccentricity of bearing on the vertical rail or the cant of the canted rail, a total eccentricity of 0.85 to 1.20 in. on the 130-lb. rail. A slope of 1 in 10 from the center of bearing of wheel on rail to the mid-point of the base of the tie-plate, or a little less, would appear a satisfactory value to be tested by experience.

Variations in the lateral force applied to the rail will result in variable intensities of bearing pressure on the tie, which may be decreased by increasing the length of the tie-plate. Granting sufficient tie-bearing area, the length of the tie-plate may well be expressed in terms of the height of rail and thickness of tie-plate. The length of the tie-plates used with the 130-lb. rail was about one and one-half times the vertical dimension referred to. Adequacy of bearing areas and of length of plate is a very important matter.

27.—Stresses in Rail on Curved Track.—In Figs. 86 to 90 are plotted the stresses found at the inside edge and the outside edge of the base of the outer rail and the inner rail under the wheels of the locomotive and loaded freight cars on the two 7° curves of the Baltimore and Ohio Railroad, the 10° curve of the Lehigh Valley Railroad, and the 2½° and the 5° curves of the Richmond, Fredericksburg and Potomac Railroad. The stresses given are average values for, say, ten runs with a possible of forty readable observations at each edge of each rail. The observations were so distributed through the revolution of the drivers that the effect of counterbalance was entirely masked. An effect of counterbalance may be expected, of course, which will give an increased stress at certain points in the revolution of the driver.

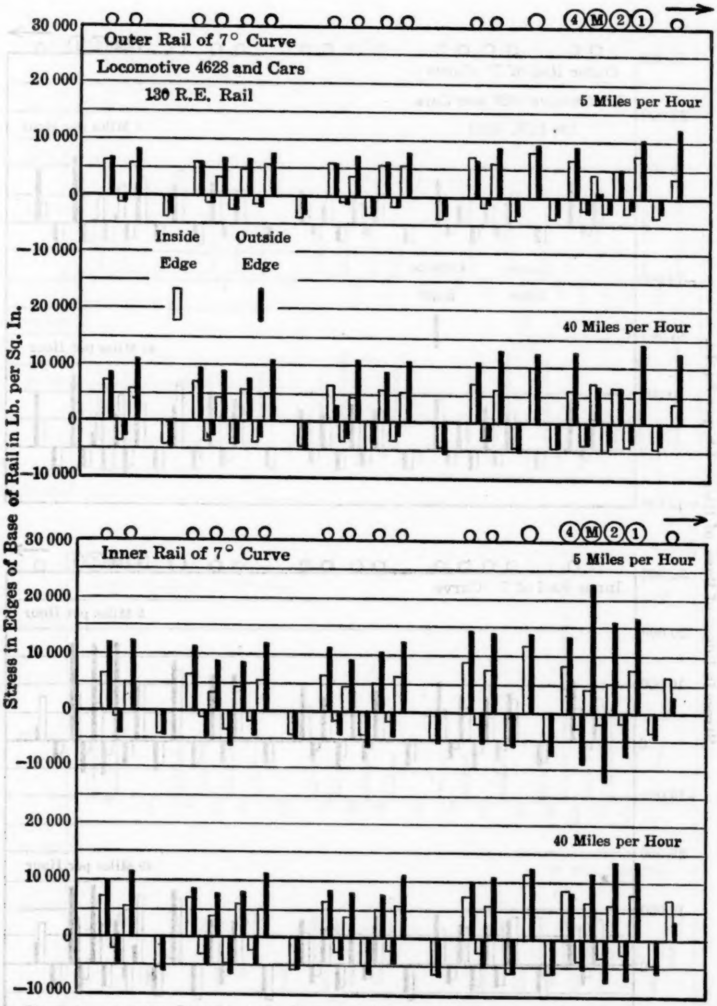


FIG. 86.—STRESS AT THE INSIDE AND OUTSIDE EDGES OF BASE OF OUTER AND INNER RAILS OF THE 7° CURVE WITH FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS. BALTIMORE AND OHIO RAILROAD.

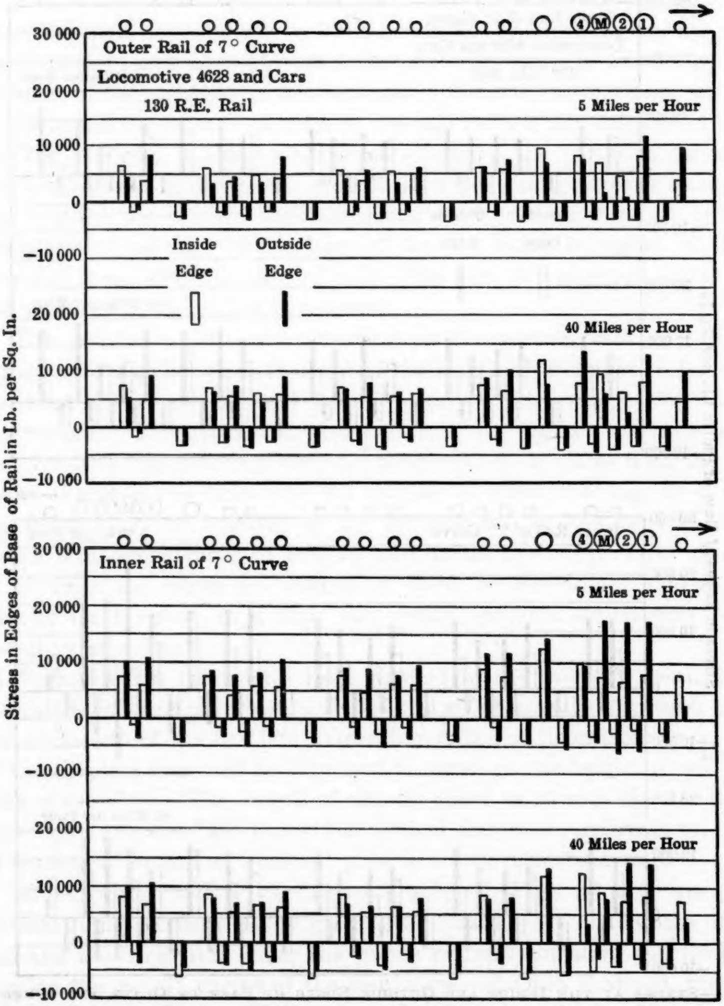


FIG. 87.—STRESS AT THE INSIDE AND OUTSIDE EDGES OF BASE OF OUTER AND INNER RAILS OF THE 7° CURVE WITH INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS. BALTIMORE AND OHIO RAILROAD.

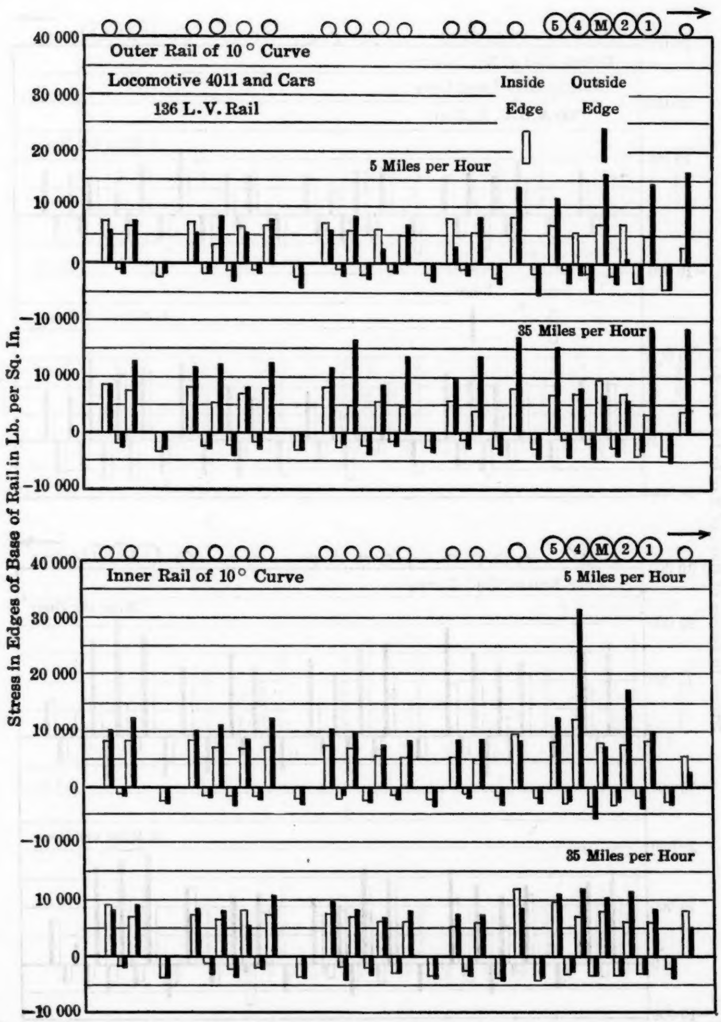


FIG. 88.—STRESS AT INSIDE AND OUTSIDE EDGES OF BASE OF OUTER AND INNER RAILS OF THE 10° CURVE WITH INCLINED TIE-PLATES. SANTA FÉ TYPE LOCOMOTIVE AND CARS, LEHIGH VALLEY RAILROAD.

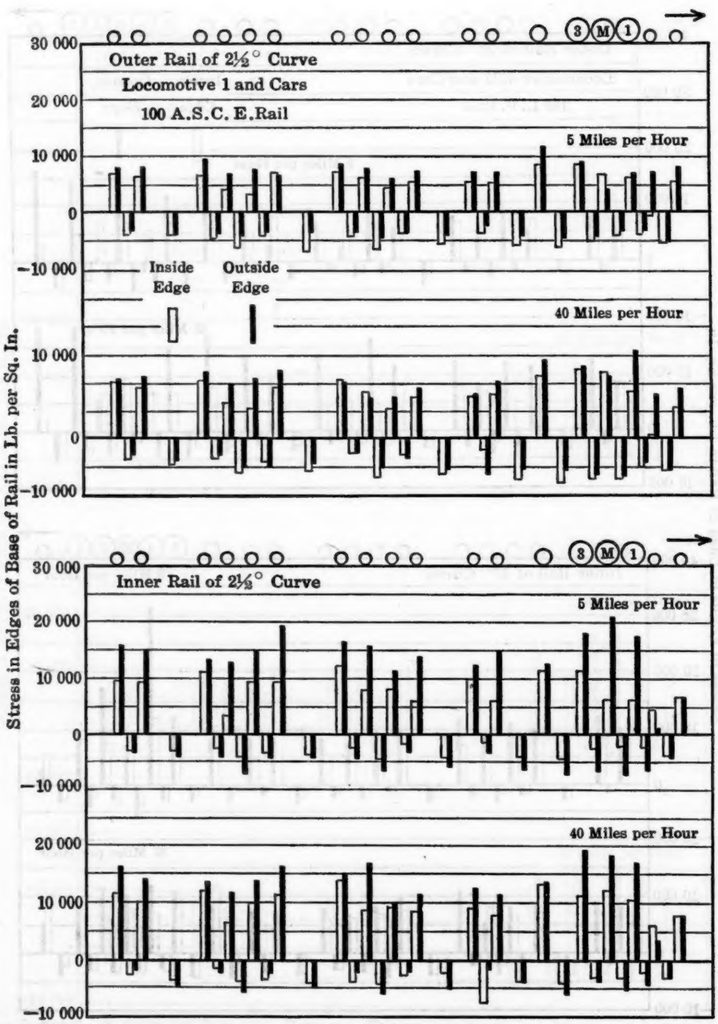


FIG. 89.—STRESS AT INSIDE AND OUTSIDE EDGES OF BASE OF OUTER AND INNER RAILS OF THE 2½° CURVE WITH FLAT TIE-PLATES. PACIFIC TYPE LOCOMOTIVE AND CARS, RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD.

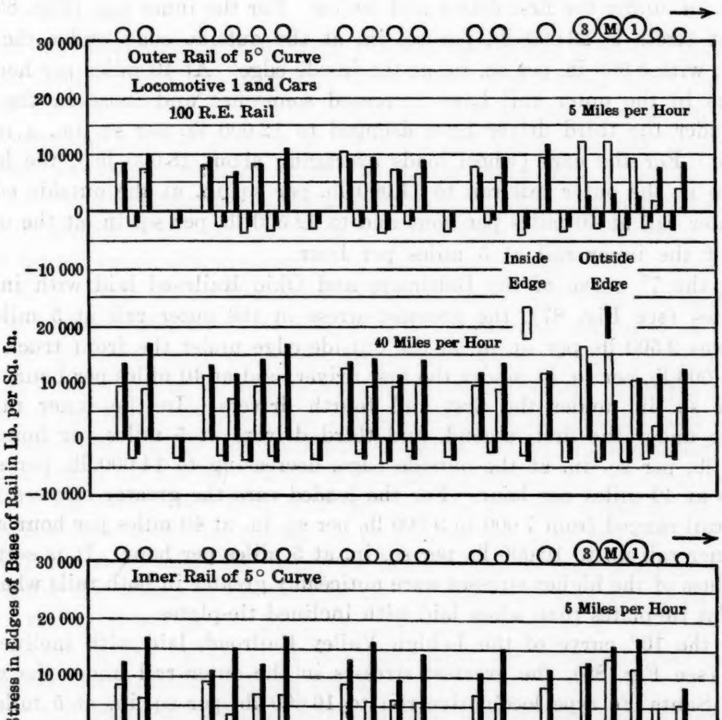


FIG. 90.—STRESS AT INSIDE AND OUTSIDE EDGES OF BASE OF OUTER AND INNER RAILS OF THE 5° CURVE WITH INCLINED TIE-PLATES. PACIFIC TYPE LOCOMOTIVE AND CARS, RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD.

On the 7° curve of the Baltimore and Ohio Railroad laid with flat tie-plates, it is seen from Fig. 86 that the highest stress in the outer rail under the wheels of the Mikado type locomotive at 5 miles per hour is 12 000 lb. per sq. in., found in the outside edge under the front truck wheel, and 10 000 lb. per sq. in. under the first driver and trailer. For the inner rail (Fig. 86), the highest stress is 23 000 lb. per sq. in. at the outside edge under the third driver, with 4 500 lb. per sq. in. at the inside edge. At 40 miles per hour, the stresses in the outer rail have increased somewhat and those in the inner rail under the third driver have dropped to 12 000 lb. per sq. in., a marked change. For the cars (wheel loads averaging about 18 000 lb.), the highest stresses in the outer rail ran to 11 000 lb. per sq. in. at the outside edge of the outer rail at 40 miles per hour and to 12 500 lb. per sq. in. at the outside edge of the inner rail at 5 miles per hour.

On the 7° curve of the Baltimore and Ohio Railroad laid with inclined tie-plates (see Fig. 87), the greatest stress in the outer rail at 5 miles per hour was 9 500 lb. per sq. in. at the outside edge under the front truck wheel and 11 000 lb. per sq. in. under the first driver, and at 40 miles per hour, 12 500 lb. per sq. in. under the first and fourth drivers. In the inner rail the stresses under the first, second, and third drivers at 5 miles per hour were 17 500 lb. per sq. in. at the outside edge, decreasing to 14 000 lb. per sq. in. or less at 40 miles per hour. For the loaded cars the greater stresses in the outer rail ranged from 7 000 to 9 000 lb. per sq. in. at 40 miles per hour and in the inner rail up to 10 000 lb. per sq. in. at 5 miles per hour. It is seen that the values of the higher stresses were noticeably greater in both rails when laid with flat tie-plates than when laid with inclined tie-plates.

On the 10° curve of the Lehigh Valley Railroad, laid with inclined tie-plates (see Fig. 88), the greatest stresses in the outer rail under the wheels of the Santa Fé type locomotive ran to 16 000 lb. per sq. in. at 5 miles per hour and to 19 000 lb. per sq. in. at 35 miles per hour, with considerable variation for the different drivers. In the inner rail the greatest stress was 31 500 lb. per sq. in. at the outside edge under the fourth driver at 5 miles per hour, with 12 000 lb. per sq. in. at the inside edge. These stresses indicate not only high lateral bending force at this driver, but show that the vertical load transmitted by this driver under the condition was far greater than the nominal load, a result that agrees with tests given in the third progress report. At the speed of 5 miles per hour, the head of the inner rail moved outwardly under the fourth driver 0.21 in., a considerable movement for so stiff a rail, especially as it occurred in a short length. At 35 miles per hour the stresses under this driver decreased to 12 000 and 7 000 lb. per sq. in. at the two edges of the rail, and there seemed to be less difficulty in its traversing the curve. The worn section of the inner rail probably contributed to the increase in mean stress in base of rail by perhaps 5%, but the effect on lateral bending was, of course, much smaller. The loaded cars (wheel load of 24 000 lb.) gave stresses in the outside edge of the outer rail up to 8 000 lb. per sq. in. at 5 miles per hour and to 12 000 lb. per sq. in. at 35 miles per hour, with a stress of 16 500 lb. per sq. in. under one wheel, the stresses being

generally higher under the first wheel of a truck. In the inner rail, the stress at the outside edge was generally the higher, and ranged to 12 500 lb. per sq. in. at 5 miles per hour and to 10 000 and 11 000 lb. per sq. in. at 35 miles per hour. In making comparison it should be borne in mind that the wheel load of these cars was 33% greater than those used on the Baltimore and Ohio Railroad.

On the 2½° curve of the Richmond, Fredericksburg and Potomac Railroad, laid with flat tie-plates (see Fig. 89), the stresses in the outer rail under the wheels of the locomotive and cars at 5 miles per hour are less than on a straight track, becoming a little greater at 40 miles per hour. In the inner rail the stresses at the outside edge under the drivers range from 17 000 to 22 000 lb. per sq. in., becoming less at 40 miles per hour. For the loaded cars the stresses in the inner rail are considerably higher than on straight track. This rail was canted outwardly, presumably by the action of traffic.

On the 5° curve of the Richmond, Fredericksburg and Potomac Railroad, laid with inclined tie-plates (see Fig. 90), the stresses in the outer rail were highest at the inside edge under the second and third drivers, 12 000 lb. per sq. in. at 5 miles per hour, and 15 500 and 16 500 lb. per sq. in., respectively, at 40 miles per hour. In the inner rail the stress at the outside edge under the first driver at 5 miles per hour was 17 000 lb. per sq. in., a smaller value than that found on the 2½° curve; it became somewhat less at 40 miles per hour. For the cars (wheel load of 25 000 lb.), the greatest stress in the outer rail was less than 10 000 lb. per sq. in. at 5 miles per hour and ran to 16 500 lb. per sq. in. at 40 miles per hour, the latter being as high a stress as was found under the wheels of the locomotive. In the inner rail the stress under the cars ran to 17 000 lb. per sq. in. at the outside edge at 5 miles per hour and to 13 000 lb. per sq. in. at 40 miles per hour, the greatest stress usually being under the first wheel of the truck.

Above and below the average stress at each edge of the base of rail here reported, there was the same range or belt of stresses as have been described in other tests.

28.—Vertical Bending Stresses on Curved Track.—As before, the mean stress in base of rail is taken as the vertical bending stress, or better, the stress in a line normal to the transverse inclination of the track. For the different speeds and the different cants the mean stress may not represent the normal bending stress exactly, but usually for the conditions of the tests it will not differ more than 1% from the correct results. The vertical bending stresses (the averages for all the runs) for the five curves are recorded in Figs. 91 to 95. For convenience, the curves are shown as curving to the right although all but the 2½° curve to the left with respect to the direction of the runs.

On the 7° curve of the Baltimore and Ohio Railroad laid with flat tie-plates, the highest vertical bending stress in the outer rail is under the first driver of the Mikado type locomotive, 10 000 lb. per sq. in. at 40 miles per hour; in the inner rail the highest stresses at 5 miles per hour are 13 600 lb. per sq. in. under the third driver and 13 200 lb. per sq. in. under the trailer,

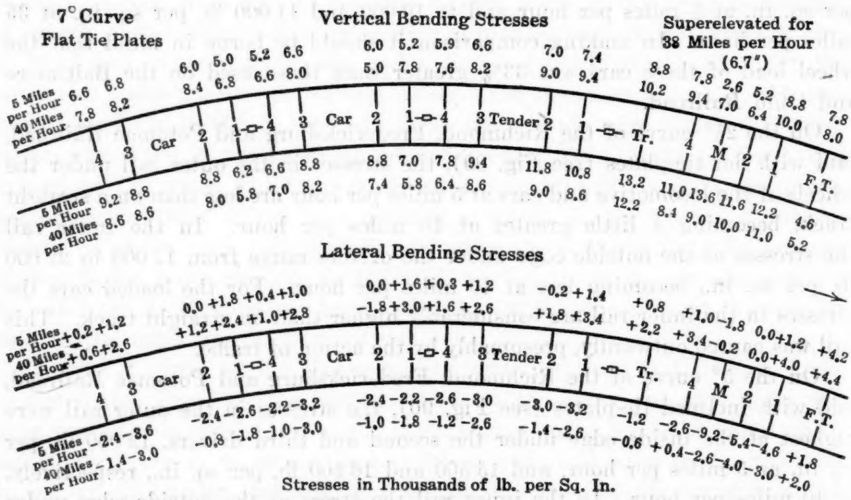


FIG. 91.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF THE 7° CURVE WITH FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS. BALTIMORE AND OHIO RAILROAD.

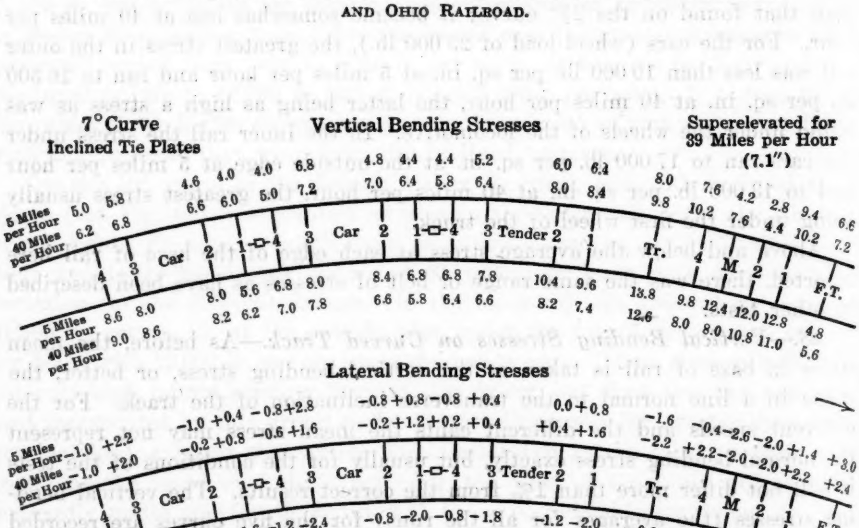


FIG. 92.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF THE 7° CURVE WITH INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS. BALTIMORE AND OHIO RAILROAD.

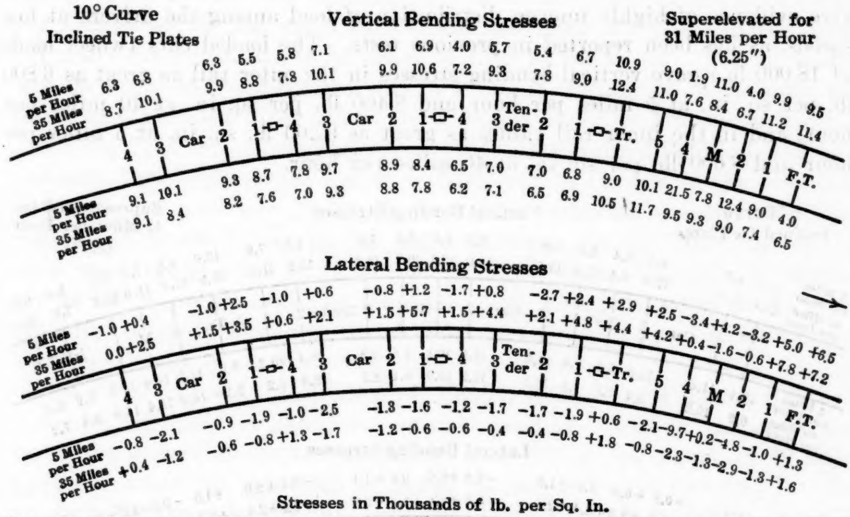


FIG. 93.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF THE 10° CURVE WITH INCLINED TIE-PLATES. SANTA FÉ TYPE LOCOMOTIVE AND CARS. LEHIGH VALLEY RAILROAD.

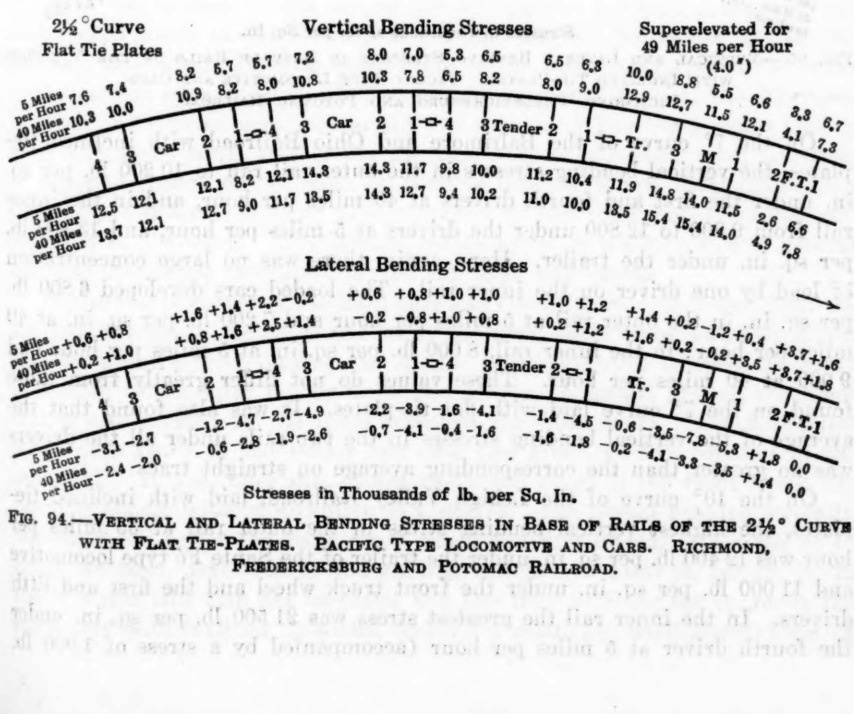


FIG. 94.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF THE 2½° CURVE WITH FLAT TIE-PLATES. PACIFIC TYPE LOCOMOTIVE AND CARS. RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD.

and at 40 miles per hour 11 000 and 12 200 lb. per sq. in., respectively, under the same wheels. The stresses in the inner rail at 5 miles per hour do not give evidence of highly uneven distribution of load among the drivers at low speeds, as has been reported in previous tests. The loaded cars (wheel loads of 18 000 lb.) gave vertical bending stresses in the outer rail as great as 6 800 lb. per sq. in. at 5 miles per hour and 8 400 lb. per sq. in. at 40 miles per hour, and in the inner rail values as great as 9 200 lb. sq. in. at 5 miles per hour and 8 600 lb. per sq. in. at 40 miles per hour.

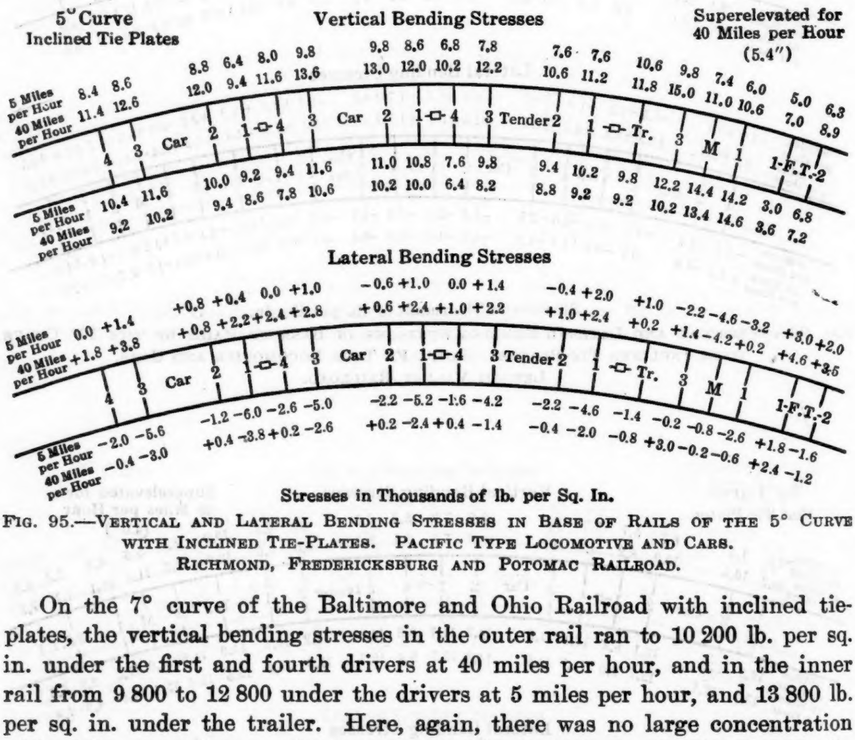


FIG. 95.—VERTICAL AND LATERAL BENDING STRESSES IN BASE OF RAILS OF THE 5° CURVE WITH INCLINED TIE-PLATES. PACIFIC TYPE LOCOMOTIVE AND CARS. RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD.

On the 7° curve of the Baltimore and Ohio Railroad with inclined tie-plates, the vertical bending stresses in the outer rail ran to 10 200 lb. per sq. in. under the first and fourth drivers at 40 miles per hour, and in the inner rail from 9 800 to 12 800 under the drivers at 5 miles per hour, and 13 800 lb. per sq. in. under the trailer. Here, again, there was no large concentration of load by one driver on the inner rail. The loaded cars developed 6 800 lb. per sq. in. in the outer rail at 5 miles per hour and 7 200 lb. per sq. in. at 40 miles per hour; in the inner rail, 8 000 lb. per sq. in. at 5 miles per hour and 9 000 at 40 miles per hour. These values do not differ greatly from those found on the 7° curve laid with flat tie-plates. It was also found that the average of the vertical bending stresses in the two rails under all the drivers was no greater than the corresponding average on straight track.

On the 10° curve of the Lehigh Valley Railroad, laid with inclined tie-plates, the highest vertical bending stress in the outer rail at 35 miles per hour was 12 400 lb. per sq. in. under the trailer of the Sante Fé type locomotive and 11 000 lb. per sq. in. under the front truck wheel and the first and fifth drivers. In the inner rail the greatest stress was 21 500 lb. per sq. in. under the fourth driver at 5 miles per hour (accompanied by a stress of 1 900 lb.

per sq. in. in the outer rail), showing a severe concentration of load on this inner driver, which must result in injury to the inner rail, results which have been found elsewhere with this type of locomotive when the third driver was flangeless. At the speed of 35 miles per hour the stress in the inner rail under the fourth driver decreased to 9 500 lb. per sq. in. For the loaded cars the stress in the outer rail ran to 10 600 lb. per sq. in. at 35 miles per hour and that in the inner rail to 10 100 lb. per sq. in. at 5 miles per hour.

On the 2½° curve of the Richmond, Fredericksburg and Potomac Railroad, laid with flat tie-plates, the higher vertical bending stresses in the outer rail at 5 miles per hour were 13 000 lb. per sq. in. under the wheels of the locomotive and 11 000 lb. per sq. in. under the cars. In the inner rail at the same speed, the greatest stress was 14 800 lb. per sq. in. under the locomotive and 14 300 lb. per sq. in. under the cars. Little change in stress in the inner rail with change in speed was found under locomotive or cars, but an increase with increase in speed was found in the outer rail.

On the 5° curve of the Richmond, Fredericksburg and Potomac Railroad, laid with inclined tie-plates, the vertical bending stress in the outer rail under the drivers of the locomotive ran to 15 000 lb. per sq. in. at 40 miles per hour and in the inner rail to 14 500 lb. per sq. in. at both speeds. For the cars, the stress in the outer rail was 9 400 to 13 600 lb. per sq. in at 40 miles per hour and in the inner rail, 9 200 to 11 600 lb. per sq. in. at 5 miles per hour. The stresses found in the 5° curve were not greater than those in the 2½° curve.

29.—Lateral Bending Stresses on Curved Track.—In Figs. 91 to 95 are recorded the average values of the lateral bending stresses in base of rail under each wheel of the locomotive and cars for the five curves at the two speeds. The lateral bending stress given is the difference between the stress at the outside edge and the mean stress in base of rail (half of the difference between the stresses measured at the two edges). When the positive sign is used, the stress tends to increase the curvature of the rail; when negative, it tends to straighten the rail. A positive bending stress, then, indicates a tendency to deflect the outer rail outwardly of the track and the inner rail inwardly of the track. This flexural action exists in addition to any general outward or inward movement of the rail that may occur.

In the two 7° curves of the Baltimore and Ohio Railroad, it is seen from the positive stresses in the outer rail under the front truck wheel and first driver that the guiding force making the Mikado type locomotive traverse the curve is applied to the outer rail by these two wheels. A lateral pull on the inner rail is also exerted by the inner wheel of the front truck. The first outer wheel of a truck of the tender and cars also gives positive bending in the outer rail. In the inner rail, except under the wheel of the front truck, the bending almost without exception is negative. The lateral bending stress in the outer rail under the front truck wheel and first driver, at a speed of 40 miles per hour, developed by the guiding action, was 4 000 lb. per sq. in. in the 7° curve laid with flat tie-plates and less than 3 000 lb. per sq. in. in the curve laid with inclined tie-plates, which are relatively small lateral bending

stresses. The outward thrust of the drivers on the inner rail develops lateral bending stresses of 5 000 and 9 000 lb. per sq. in. in the track laid with flat tie-plates and 5 000 lb. per sq. in. in that laid with inclined tie-plates. For both outer and inner rail the lateral bending stresses under the wheels of the cars are less in the track laid with inclined tie-plates than in that laid with flat tie-plates. It is seen that, with all other conditions the same, the lateral bending stresses in both rails were smaller for the locomotive and cars on the curve with the rail canted. Except for one driver on one curve, the lateral bending stresses were well distributed among the wheels, and the train traversed the curve seemingly without difficulty.

The lateral bending stresses in the outer rail of the 10° curve of the Lehigh Valley Railroad under the front truck wheel and first driver of the Santa Fé type locomotive, at 35 miles per hour, were 7 200 and 7 800 lb. per sq. in., respectively, values which are about two-thirds of the corresponding vertical bending stress. At these wheels the head of the rail moved outward 0.12 in. at the speed of 35 miles per hour. In the inner rail under the fourth driver, at a speed of 5 miles per hour, a high lateral bending stress was found, a characteristic of this type of locomotive when the third driver is flangeless, 9 700 lb. per sq. in., tending to strengthen the rail and indicating a strong outward push on it at this point, the rail moving outward 0.2 in. This lateral bending stress was nearly one-half as much as the extremely high vertical bending stress found under this driver. At 35 miles per hour the lateral bending stress was only 2 300 lb. per sq. in., about one-fourth the vertical bending stress found at this speed. The lateral bending stresses in the outer rail under the first wheel of a truck of the tender and cars indicate that these wheels guide the truck around the curve. There is considerable variation in the lateral bending stresses under these wheels, the range being from 2 000 to 5 700 lb. per sq. in., at 35 miles per hour, all being high with respect to the vertical bending stresses at the same points. The lateral bending stresses in the inner rail were low at both speeds and generally indicated negative bending. No comparison of lateral bending stresses with rail laid with flat tie-plates can be made.

On the 2½° curve of the Richmond, Fredericksburg and Potomac Railroad, laid with flat tie-plates, the outer truck wheels give positive bending at both speeds, as does also the first driver of the Pacific type locomotive at 40 miles per hour, the lateral bending stresses being less than 4 000 lb. per sq. in. In the inner rail, there is a lateral bending stress of 7 800 lb. per sq. in. under the main driver, at 5 miles per hour, more than one-half as great as the vertical bending stress. Although the stress is decreased to 3 300 lb. per sq. in., at 40 miles per hour, it is evident that this wheel exercises a strong outward pressure against the inner rail and thus may have some influence in producing the outward canting of this rail. Under the cars the lateral bending stresses in the outer rail are small, but there are indications that the trucks were tending to run obliquely toward the inner rail. The lateral bending in the inner rail under the cars is inward, but the stresses under the front wheel of a truck are high at 5 miles per hour, reaching 5 000 lb. per sq. in.; at 40

miles per hour the stresses at these points are small, but the values are larger than the corresponding ones in the outer rail. Doubtless, part of this was due to the condition of the track at the inner rail.

On the 5° curve of the Richmond, Fredericksburg and Potomac Railroad, laid with inclined tie-plates, the outward lateral bending in the outer rail was under the front truck wheels, and evidently these wheels gave the guiding action. The lateral bending stresses were 3 500 and 4 600 lb. per sq. in., respectively, at 40 miles per hour. The lateral bending stress in the inner rail under the middle driver is very small. The lateral bending stresses in the outer rail under the tender and cars are greater for the first wheel of each truck, usually about 2 500 lb. per sq. in., at 40 miles per hour. The lateral bending stresses in the inner rail at 5 miles per hour were relatively high, ranging to 6 000 lb. per sq. in. under the first wheels of a truck. At 40 miles per hour the corresponding stresses were about one-half as much.

In Fig. 96 are plotted the values of the lateral bending moments developed in the outer and inner rail at and between wheels on the two 7° curves of the Baltimore and Ohio Railroad. The values were calculated from the average lateral bending stresses and the section modulus, $\frac{I}{c}$, of the rail about a vertical axis. They represent the tendency of the wheels of the Mikado type locomotive and cars to give lateral distortion to the rail and track.

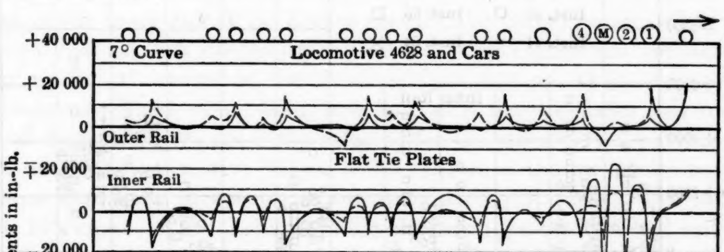


FIG. 96.—LATERAL BENDING MOMENTS IN OUTER AND INNER RAILS OF THE 7° CURVE. MIKADO TYPE LOCOMOTIVE AND CARS, BALTIMORE AND OHIO RAILROAD.

In Fig. 97, the individual lateral bending stresses at the several instruments are plotted for a speed of 40 miles per hour for the runs on the 7° curve of the Baltimore and Ohio Railroad, laid with inclined tie-plates, values under the wheels of the locomotive and of the first car being shown. It is seen

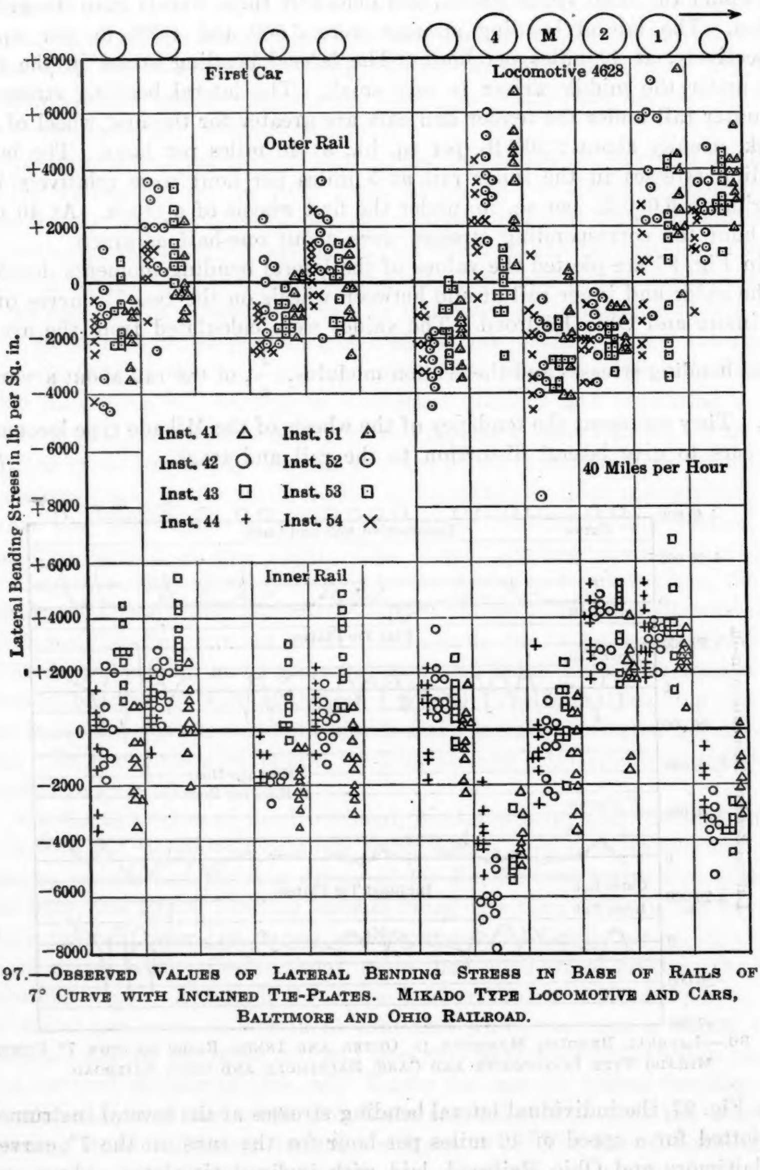


FIG. 97.—OBSERVED VALUES OF LATERAL BENDING STRESS IN BASE OF RAILS OF THE 7° CURVE WITH INCLINED TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS, BALTIMORE AND OHIO RAILROAD.

that there is a range of values for each instrument, one point on the track giving different values from another, yet the variation being much the same for different instruments and different wheels. The individual drivers give quite different results. It is evident that different runs give different results, and that the variation in values is due both to differences in the action of the train and to difference in the track from point to point.

30.—Movement of the Rail on Curved Track.—In Fig. 98 are given the measurements of tilting of the rail on the two 7° curves of the Baltimore and Ohio Railroad under the action of the locomotive and cars, and also those on the 10° curve of the Lehigh Valley Railroad and the 5° curve of the Richmond, Fredericksburg and Potomac Railroad, the tilting of the head of the rail with respect to the base being measured as already described for the tests on straight track. The values given are the averages of two or three runs at the same point.

It is seen that the outer rail tilts outwardly very little under the wheel of the locomotive that guides the driver group. It is also seen that the inner rail generally tilts outwardly under one or more of the intermediate drivers. In general, for the other wheels of the locomotive and the wheels of the tender and cars, not only for the tests shown in the diagrams but also for the other tests, both the outer rail and the inner rail tilt inwardly and the inward tilting is about the same in the outer rail for both speeds and is greater in the inner rail at the higher speed. The tilting movement is usually less than 0.05 in., but ranges up to 0.10 in. The nature and amount of the tilting on two 1° curves of the Reading Company on which measurements of movement of rail were taken, one laid with flat tie-plates and one with inclined tie-plates, were much the same as for the other curves referred to, the action under the locomotive wheels being typical of the movement on curves.

There were some exceptions to the statements just made. The inner rail of the 10° curve of the Lehigh Valley Railroad generally tilted outwardly, markedly so under the drivers of the locomotive. It will be recalled that this inner rail was a worn rail relaid from the outer rail of the curve. The outer rail of the 2½° curve of the Richmond, Fredericksburg and Potomac Railroad tilted outwardly under both the locomotive and cars and the tilt of the inner rail under the tender and cars was slight, sometimes inwardly and sometimes outwardly. Likewise, the tilt of the inner rail of the 5° curve of the Richmond, Fredericksburg and Potomac Railroad was slight and variable in direction.

In Fig. 99 are given the measurements of the lateral movement of the head of the rail for the same four curves. An outward movement of the outer rail under the wheels which act to guide the driver group, is shown, and generally an inward movement under the other wheels. For the inner rail the movement is generally inward, except for the intermediate drivers which usually give an outward movement. The amount of the lateral movement of the head of the inner rail ranges from — 0.15 in. to + 0.20 in. The highest lateral movement of the inner rail occurred under the fourth driver of the Santa Fé type locomotive, a movement that took place almost entirely between the third

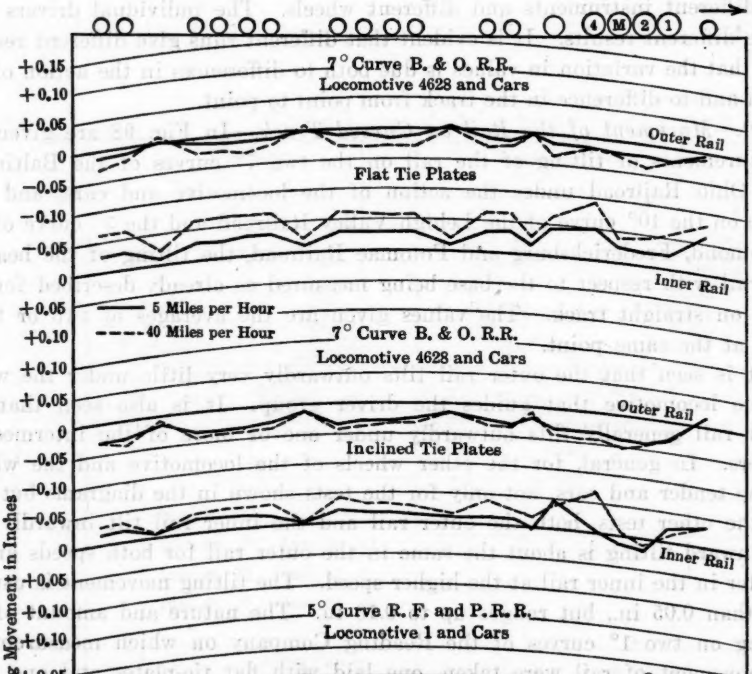


FIG. 98.—TILTING MOVEMENT OF RAILS ON CURVED TRACK. BALTIMORE AND OHIO RAILROAD, LEHIGH VALLEY RAILROAD, AND RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD.

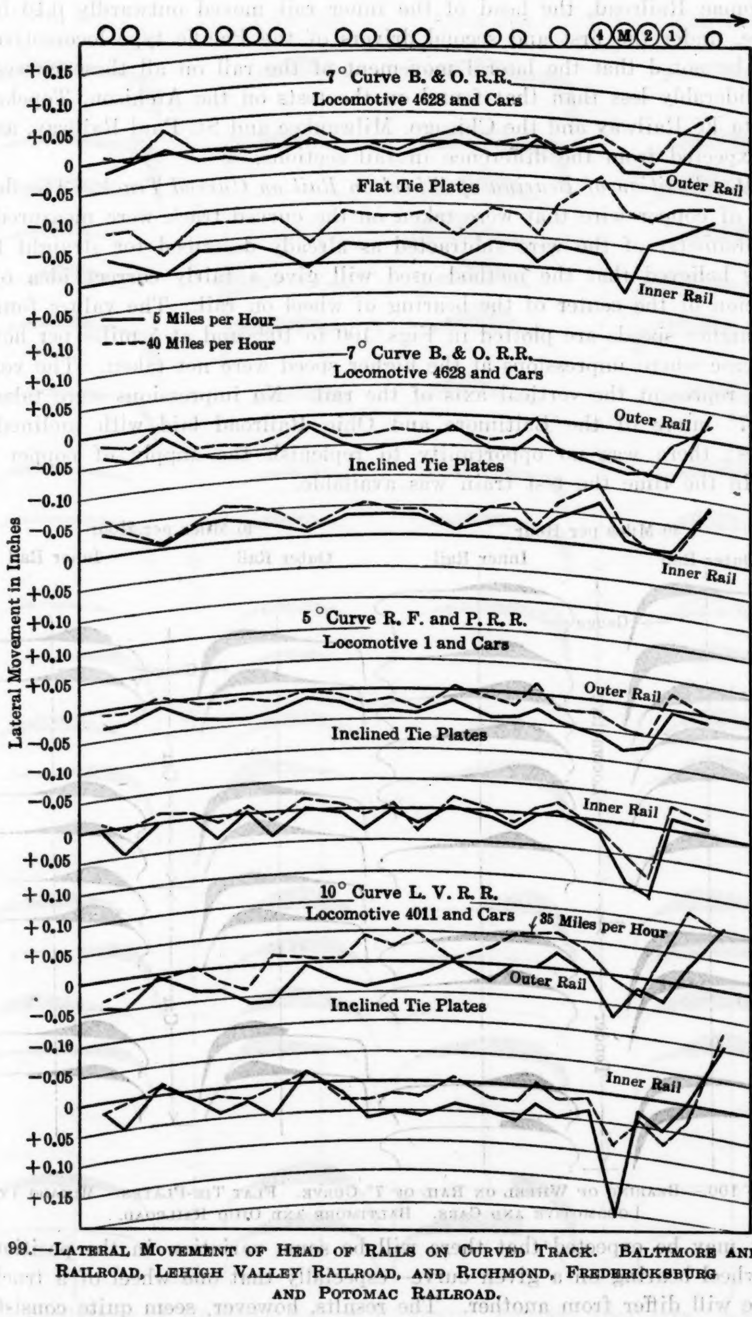


FIG. 99.—LATERAL MOVEMENT OF HEAD OF RAILS ON CURVED TRACK. BALTIMORE AND OHIO RAILROAD, LEHIGH VALLEY RAILROAD, AND RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD.

and fifth drivers. On the two curves of the Richmond, Fredericksburg and Potomac Railroad, the head of the inner rail moved outwardly 0.10 in., or more, under the first and second drivers of the Pacific type locomotive. It will be noted that the lateral movement of the rail on all these curves was considerably less than that found on the tests on the Atchison, Topeka and Santa Fé Railway and the Chicago, Milwaukee and St. Paul Railway, as may be expected from the difference in rail sections.

31.—Position of Bearing of Wheel on Rail on Curved Track.—The flattenings of copper wire that were taken on the curved track were measured and the diameter of the wire subtracted as already described for straight track. It is believed that the method used will give a fairly correct idea of the position of the center of the bearing of wheel on rail. The values found at the higher speeds are plotted in Figs. 100 to 102, and at 5 miles per hour in the case where impressions at the higher speed were not taken. The vertical lines represent the vertical axis of the rail. No impressions were taken on the 7° curve of the Baltimore and Ohio Railroad laid with inclined tie-plates; there was no opportunity to replenish the supply of copper wire within the time the test train was available.

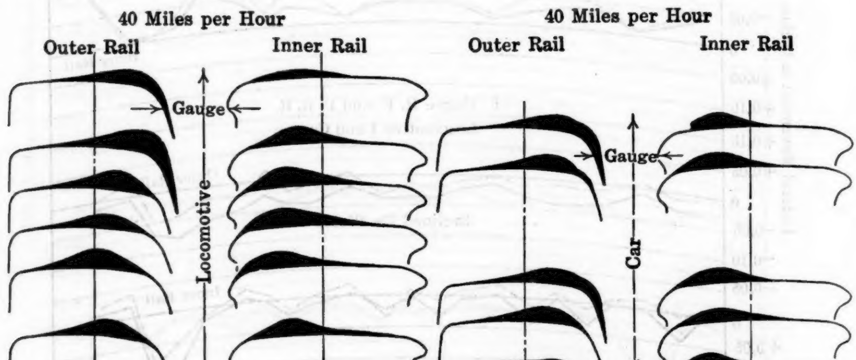


FIG. 100.—BEARING OF WHEEL ON RAIL OF 7° CURVE. FLAT TIE-PLATES. MIKADO TYPE LOCOMOTIVE AND CARS. BALTIMORE AND OHIO RAILROAD.

It may be expected that there will be some variation in the position of the wheel bearing on a given curve—especially that one wheel of a truck or frame will differ from another. The results, however, seem quite consistent. Describing the position of the center of bearing in terms of the width of the

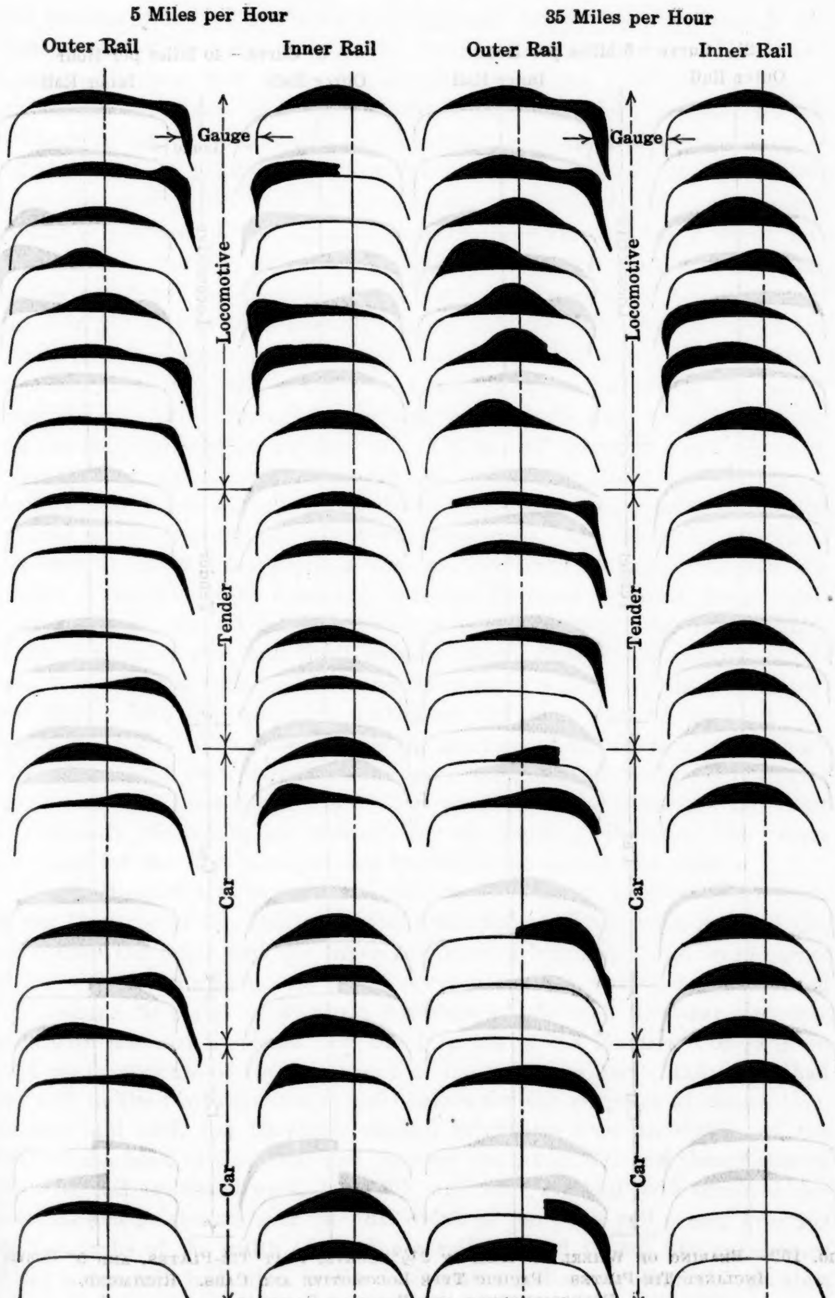


FIG. 101.—BEARING OF WHEEL ON RAIL OF 10° CURVE. INCLINED TIE-PLATES. SANTA FÉ TYPE LOCOMOTIVE AND CARS. LEHIGH VALLEY RAILROAD.

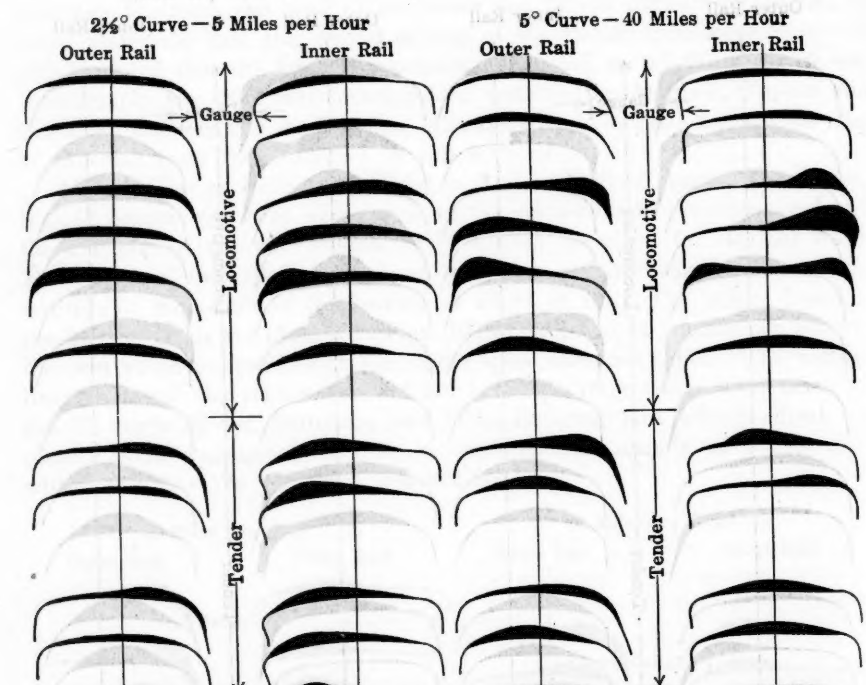


FIG. 102.—BEARING OF WHEEL ON RAIL OF 2½° CURVE, FLAT TIE-PLATES, AND 5° CURVE, INCLINED TIE-PLATES. PACIFIC TYPE LOCOMOTIVE AND CARS. RICHMOND, FREDERICKSBURG AND POTOMAC RAILROAD.

flatter portion of the top of the head (this is not considering the curves at the corners) and counting always from the gauge side of the rail, a study of the diagrams will show that the center of pressure on the outer rail of the 7° curve of the Baltimore and Ohio Railroad laid with flat tie-plates for all the wheels of the locomotive and cars usually lies at from one-fourth to one-third the width of head, except that for the fourth driver 0.45 will denote the position, and on the inner rail one-fourth to one-third will also represent the relative position of the center of pressure, both rails having a slight outward cant. On the 10° curve of the Lehigh Valley Railroad, at a speed of 35 miles per hour, the position of the center of bearing on the outer rail (cant 1 in 36) was at from one-third to one-half the width for all except the third driver (flangeless), which was at three-fourths the width. On the inner rail (cant, 1 in 27), the position was usually near four-tenths the width except for the third driver (flangeless), which was 0.55, counting in all cases the full width of an unworn rail. On the 2½° curve of the Richmond, Fredericksburg and Potomac Railroad, at 5 miles per hour, the position on the outer rail (cant slightly outward) was one-third to one-half, except for the third driver (worn tire), which was two-thirds. On the inner rail (outward cant, 1 in 32), the position was at one-fourth to one-third. On the 5° curve of the Richmond, Fredericksburg and Potomac Railroad, the position on the outer rail (cant, 1 in 26) was close to one-half, except that it was three-fourths for the second and third drivers (worn tires). On the inner rail (cant, 1 in 42), the position was one-half or more, except for the second and third drivers with the worn tires resting on the outer part of the head. In general, there was little difference in the position of the bearing at the high and the low speed, even though at one speed the flange of the wheel might be closer to one rail than to the other. The position of the wheels on the outer rail generally showed flange pressure for the guiding wheels of the driver group and for the first wheel of the trucks of the tender and cars.

The position of the wear of the rail on the curved track is of interest. On the 7° curve of the Baltimore and Ohio Railroad laid with inclined tie-plates, both the outer and the inner rail showed brightness and wear across the full width of the head. As the 7° curve with flat tie-plates had been used with inclined tie-plates to within a few days of the test, the wear was that found with inclined tie-plates. Of the 1° curve of the Reading Company on which measurements of the movement of the rail were made, and which had been laid in the two ways two months before for the purposes of observation, the part laid with flat tie-plates showed brightness over two-thirds of the width of the head of the outer rail (inward cant, 1 in 60) and three-fifths of the inner rail (outward cant, 1 in 30), and the part laid with inclined tie-plates showed brightness over the full width of the outer rail (cant, 1 in 15) and five-sixths of the width of the inner rail (cant, 1 in 24), the appearence of the outer rail laid with inclined tie-plates indicating that the greatest bearing came silghtly outside the middle of the head. On the 10° curve of the Lehigh Valley Railroad, laid with inclined tie-plates, the brightness was over nearly the full width of the outer rail (cant, 1 in 36) and the greatest possible width of the worn inner rail (cant, 1 in 27). The outside of the

head of the inner rail (re-laid from an outer rail) was so worn that the center of the whole bearing area of the head was necessarily to one side of the center line of the web. On the 5° curve of the Richmond, Fredericksburg and Potomac Railroad, laid with inclined tie-plates, the brightness extended over three-fourths of the width of the outer rail (cant, 1 in 26), the gauge side of the head being somewhat worn, and on the inner rail (cant, 1 in 42), the maximum wear was a little inside the middle of the head.

Estimates were made of the position of the center of wheel bearing for the various wheels of the locomotive, tender, and cars, and an average value of the distance of these centers from the middle of the rail head was found. The average position of the center of bearing shown by the impressions on copper wire was calculated, but some consideration was given to the position of the brightness of the rail, and when the wire impressions were not available the other observations were used. The values so estimated are given in Column (3) of Table 17; they are considered to be more accurate than the preceding general statements. In making up Table 17, the position of the wheels and drivers most actively engaged in forming the guiding action was not taken into consideration.

32.—Discussion of Canted Rail and of Unsymmetrical Tie-Plates on Curved Track.—The study of the data in the preceding articles has shown that the center of the bearing of the wheel on the rails of the curved track tested is more nearly at the middle of the head of the rail when the rail is canted inwardly than when it is vertical. It is apparent from the tests that a cant of 1 in 20 will give a satisfactory bearing position for the wheels of the locomotives and cars used in the tests. The position of the wheel bearing does not differ greatly whether or not the flange of the wheel runs close to the rail, except in the case of locomotives having worn driver tires. In the case of rail that is normal or nearly normal to the transverse section of the track, the position of the center of wheel bearing ranged from 0.25 to 0.60 in. inwardly of the middle of the head of the rail, 0.50 in. being a common value. It is also apparent that in curved track as in straight track a cant of 1 in 20 will bring the center of the wheel bearing close to the middle of the head of the rail. The appearance of brightness over the head of the rail and the distribution of the wear also favored the canted rail. This is well shown in the 1° curve of the Reading Company, laid for the purpose of observation on the effect of canting, where the part laid with inclined tie-plates showed brightness over the full width of the head and that laid with flat tie-plates showed brightness only over two-thirds and three-fifths of the width. It was also found that on the two 7° curves of the Baltimore and Ohio Railroad the lateral bending stresses in the rail laid with inclined tie-plates were somewhat less than those found when flat tie-plates were used. It is not apparent that canting the rail will have any marked effect on the rapidity of wear of the gauge side of the outer rail by the flanges of wheels which bear against it and guide a truck or group of wheels in traversing the curve, except as the decrease in lateral bending stresses may show a reduction in flange pressures. Taken altogether the tests indicate that for curved track the canting of the rail conduces to a wider distribution of the bearing pressure on the head of the

rail and a more favorable distribution of wear, and thus may be expected to improve the conditions for track maintenance.

It is not apparent that canting of the rail has any marked effect on the tilting of the rail under load, or on changes in gauge by traffic, or on tie cutting. It should also be noted that the variation in lateral bending stresses for different runs and different points along the track is present both in track having vertical rail and in track having canted rail and to approximately the same degree.

Except for certain wheels of the locomotive, the action of the wheels was to tilt both outer and inner rail inwardly of the track a small amount, thus narrowing the gauge at the wheels. In the same way for the same wheels there was a movement of both rails inwardly of the track, also tending to narrow the gauge under these wheels. This condition was found to exist with both vertical and canted rail. This inward tilting and lateral movement of the rail may be attributed to the position of the tie-plate. If it is generally characteristic of such track, the inference may be drawn that for resisting the lateral forces present when this tilting takes place the eccentricity of the tie-plate is too great. Other circumstances point to another conclusion, at least for the inner rail.

It will be noted that in general the cant found on the curved track was less than that which would be expected with the tie-plates used. The inclined tie-plates had an inclination of 1 in 20—generally the rail was found to be canted less than this. The rail laid with flat tie-plates was usually found to be canted outwardly. It is reasonable to assume that when the rail was laid its position approached that which would be expected for the tie-plates used. This being true, it would seem that curved track tends to lose its cant under traffic, or to gain an outward cant, and thus to widen the gauge slightly. This change is opposite to that found in straight track, where the tendency is toward an increase in cant. It is also contrary to the tilting action found on curved track under most of the wheels. It seems evident, however, that there is a tendency for the rails of curved track to lose their cant or to become canted outwardly.

The reduction in cant may be due in part to the large outward pressures exerted on the outer rail by the flanges of the guiding wheels of the locomotive and on the inner rail by the intermediate drivers that resist the pressures exerted by the outer wheels of the locomotive. That these forces are important is borne out by the outward tilting and outward movement of the two rails under these wheels.

Another method of attack may lie in examining the probable eccentricity of application of the forces that act on curved track in somewhat the same way as was done with straight track, considering the point of application of the wheel load, the center of the tie-plate, the cant of the rail, and its superelevation, etc. It is evident from a study of the lateral bending stresses that in addition to the lateral forces developed by the guiding action of groups of wheels there are outward lateral forces acting on the rail similar in nature to those giving an outward bending in straight track; the resultant of these forces and the vertical loads may be expected to have an outward trend. Although the magnitude of these lateral forces can not be determined, it will be well to

ascertain whether the conditions compare with straight track and what the effect of speed on the conditions of super-elevated track will be.

In Table 17 are given values of the position of the wheel bearing on the rail, eccentricity of tie-plate, eccentricity due to cant of rail, eccentricity due to super-elevation, etc., for the several curves on which tests were made. Fig. 103 will help in understanding the terms used. At the speed of super-elevation the resultant of the centrifugal force and the vertical load may be considered to be normal to the transverse section of the track—parallel to the axis of the rail for flat tie-plates (*A B*, Fig. 103) and inclined to it for inclined tie-plates (*A′ B′*). At the low speeds the centrifugal force is negligible and the resultant of the vertical load and the centrifugal force may be taken to be vertical (*A C* and *A′ C′*). The nominal eccentricity then is the distance from the middle of the tie-plate to its intersection with the resultant used. This nominal resultant (*A D* and *A′ D′*) is assumed to be acting along a line joining the center of the wheel bearing and the mid-point of the tie-plate. If the rolling action of the wheel gives an outward lateral force of a magnitude such that the resultant of this and the other forces coincides with the so-called nominal resultant, the conditions would be called very favorable. If, however, under one of the speeds the resultant of the actual forces can not be expected to occupy the assumed position, it will be evident that tilting forces will be present to modify the cant of the rail. The slope of the nominal resultant may then be used as a criterion. Even for another speed, the resultant may not coincide in position with the assumed nominal resultant.

It will be seen that for the speed of super-elevation the nominal eccentricity (*B D* or *B′ D′*, Fig. 103) and the slope of the nominal resultant (tangent of angle between *A B* and *A D* or *A′ B′* and *A′ D′*), as may be expected, for both rails would not differ from those for straight track provided the position of the wheel bearing and the cant of the rail were the same. The general slope of the nominal resultant for both inner and outer rails ranges generally from 0.11 to 0.15. It would appear that at the speed of super-elevation, except for the guiding action of groups of wheels, the conditions for curved track will be much the same as for straight track.

At a slow speed it is seen that for the outer rail the line of the vertical load intersects the bottom of the tie-plate well inside its middle and that the nominal eccentricity (*C D* or *C′ D′*) is great, the values in Table 17 ranging from 0.82 to 1.78 in. The slope of the nominal resultant is correspondingly large—from 0.13 to 0.24. It is not to be expected that the outward lateral force on the rail produced by what has been called the outward rolling action of the wheels will be sufficient to give such a slope. With the actual resultant falling inside the mid-point of the tie-plate, the tendency will be for the outer rail to tilt inwardly as the load passes, except as the guiding flanges may give an outward pressure.

For the inner rail at a slow speed, the line of the vertical load intersects the bottom of the tie-plate generally not far from its mid-point, the nominal eccentricity (*C D* or *C′ D′*) given in Table 17 ranging from — 0.07 to 0.69 in. The values of the slope of the nominal resultant are very small. For a slow speed, then, the action of the traffic may be expected to tilt the inner rail outwardly, resulting in a decreased cant. This change will also be influenced

TABLE 17.—CENTER OF WHEEL BEARING, ECCENTRICITY OF TIE-PLATE, CANT OF RAIL, NOMINAL ECCENTRICITY, AND SLOPE OF NOMINAL RESULTANT ON CURVED TRACK.

(The values are given in inches, the positive direction being inward from the middle of the head of the rail. The position of the center of wheel bearing was estimated from the impressions of copper wire for the wheels of locomotive and cars when available and otherwise from the appearance of the rail and other observations. The eccentricity due to super-elevation is that distance along the tie which corresponds to the height of the rail and thickness of tie-plates for a flat tie-plate. The other terms are the same as have been used for straight track.)

Location.	Average cant of rail.	Center of wheel bearing.	Eccentricity of tie-plate.	Eccentricity due to cant of rail.	Eccentricity due to super-elevation.	At Speed of Super-Elevation.		At Slow Speed.	
						Nominal eccentricity.	Slope of nominal resultant.	Nominal eccentricity.	Slope of nominal resultant.
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Outer Rail.									
Baltimore & Ohio R. R.:									
7° Curve:									
Flat tie-plate	—1:120	0.50	0.50	—0.06	0.84	0.94	0.13	1.78	0.24
Inclined tie-plate	1:23	0.00	0.50	0.31	0.86	0.81	0.11	1.67	0.23
Reading Co.:									
1° Curve:									
Flat tie-plate	1:60	0.45	0.50	0.12	0.22	1.07	0.15	1.29	0.17
Inclined tie-plate	1:15	—0.10	0.50	0.48	0.21	0.88	0.12	1.09	0.15
Lehigh Valley R. R.:									
10° Curve:									
Inclined tie-plate	1:36	0.00	0.13	0.21	0.78	0.34	0.05	1.12	0.15
Richmond, Fredericksburg & Potomac R. R.:									
2½° Curve:									
Flat tie-plate	—1:115	0.20	0.25	—0.05	0.42	0.40	0.06	0.82	0.13
5° Curve:									
Inclined tie-plate	1:26	0.00	0.50	0.25	0.59	0.75	0.12	1.84	0.21

TABLE 17.—(*Continued*).

Location.	Average cant of rail.	Center of wheel bearing.	Eccentricity of tie-plate.	Eccentricity due to cant of rail.	Eccentricity due to super-elevation.	At Speed of Super-Elevation.		At Slow Speed.	
						Nominal eccentricity.	Slope of nominal resultant.	Nominal eccentricity.	Slope of nominal resultant.
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Inner Rail.									
Baltimore & Ohio R. R.:									
7° Curve:									
Flat tie-plate	−1:120	0.60	0.50	−0.06	−0.84	1.04	0.14	0.20	0.03
Inclined tie-plate	1:23	0.00	0.50	0.31	−0.86	0.81	0.11	−0.05	−0.01
Reading Co.:									
1° Curve;									
Flat tie-plate	−1:30	0.55	0.50	−0.25	−0.22	0.80	0.11	0.58	0.08
Inclined tie-plate	1:24	0.10	0.50	0.30	−0.21	0.90	0.12	0.69	0.10
Lehigh Valley R. R.:									
10° Curve:									
Inclined tie-plate	1:27	0.30	0.13	0.28	−0.78	0.71	0.09	−0.07	−0.01
Richmond, Fredericksburg & Potomac R. R.:									
2½° Curve:									
Flat tie-plate	−1:32	0.40	0.25	−0.20	−0.42	0.45	0.07	0.03	0.00
5° Curve:									
Inclined tie-plate	1:42	0.00	0.50	0.15	−0.59	0.65	0.10	0.06	0.01

by the strong outward pressure on the inner rail exerted by the intermediate drivers and by other wheels, as is brought out in the study of the tests. The passage of trains at slow speeds and the outward pressure of certain wheels at both low and high speeds may be taken to be the cause of the loss of cant on all the curves. It would appear that for the inner rail there would be an advantage in having a greater eccentricity of tie-plate if many trains are to be run at speeds below that corresponding to the super-elevation. Although in the tests shown in Fig. 98, the inner rail generally tilted inwardly under load, it is seen that the amount of the inward tilt at the higher speed was greater than that at the lower; if then other conditions caused an outward tilt, the effect would be the greater at the low speed.

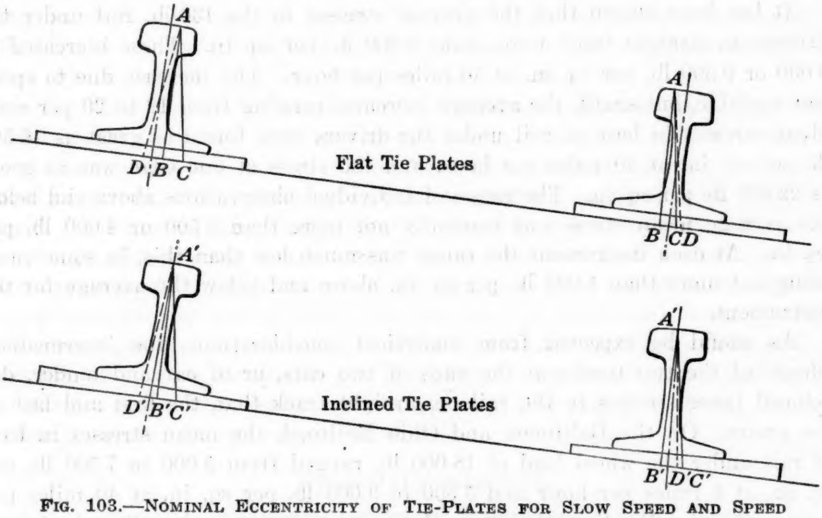


FIG. 103.—NOMINAL ECCENTRICITY OF TIE-PLATES FOR SLOW SPEED AND SPEED CORRESPONDING TO SUPER-ELEVATION.

Any tendency to changes in cant that may be made at speeds higher than those corresponding to the super-elevation will not need consideration, since the number of applications of load and their amount will be relatively small.

In the matter of the proper amount of eccentricity to be given in the design of tie-plates, it would seem that this can best be determined in practice by the results of experience on such track as that on which the tests were conducted. If it is found that the cant of the rail is decreased under service, it would be proper to increase the eccentricity of the tie-plate; if the cant shows a gain, the eccentricity may be too great. On the track used in the tests in a general way it may be said that the cant had increased on straight track and decreased on curved track.

As in the case of straight track it is of prime importance to have tie-plates of adequate length. Increasing the length will not only give increased bearing area, but the greater length will have a beneficial effect by reason of the decreased variation in pressure at the ends of the tie-plate corresponding to the variation in the direction of the resultant wheel load under the diverse conditions of traffic. In any case, the thickness of tie-plate must be sufficient for the purpose.

33.—Stresses in Heavy Rail.—The average values of the stresses in the 130-lb. rail under all the drivers of the locomotives were generally in accordance with values calculated by the analytical method given in the first progress report. Different track locations gave small differences in the average values, due partly to differences in the relative values of the positive and negative moments at and between wheels at the different locations. The values of the stresses under the individual drivers differed from the analytical results, showing a variation in the division of the load from the reported loads. As the stresses were relatively small, the variation from average values may be expected to be greater than for the lighter rail, due both to the variations in the track and to observational error.

It has been shown that the average stresses in the 130-lb. rail under the drivers on straight track were about 8 000 lb. per sq. in. These increased to 9 000 or 9 500 lb. per sq. in. at 40 miles per hour. The increase due to speed was variable, but small, the average increases ranging from 12 to 20 per cent. Mean stresses in base of rail under the drivers were found as great as 16 500 lb. per sq. in. at 40 miles per hour, and the stress at one edge was as great as 22 000 lb. per sq. in. The range of individual observations above and below the average mean stress was generally not more than 3 500 or 4 000 lb. per sq. in. At each instrument the range was much less than this, in some cases being not more than 1 000 lb. per sq. in. above and below the average for the instrument.

As would be expected from analytical considerations, the intermediate wheels of the two trucks at the ends of two cars, or of car and tender, developed lower stresses in the rail on straight track than the first and last of the group. On the Baltimore and Ohio Railroad, the mean stresses in base of rail under the wheel load of 18 000 lb. ranged from 5 000 to 7 500 lb. per sq. in. at 5 miles per hour and 5 500 to 9 000 lb. per sq. in. at 40 miles per hour. On the Reading Company the mean stresses under the wheel load of 25 000 lb. ranged from 6 000 to 10 000 lb. per sq. in. at 5 miles per hour and 6 000 to 11 000 lb. per sq. in. at 40 miles per hour. The effect of speed in the latter case was very low. It is seen that with the wheel load of 25 000 lb. the stresses in rail on straight track were nearly as great as any found under the drivers of the locomotive.

In the outer rail of the 7° curves of the Baltimore and Ohio Railroad average vertical bending stresses of 9 000 and 10 000 lb. per sq. in. were developed under a driver of the Mikado type locomotive at speeds of 5 and 40 miles per hour, respectively, and average stresses at one edge of 12 000 and 14 000 lb. per sq. in. at the two speeds. In the inner rail average vertical bending stresses of 13 500 and 11 000 lb. per sq. in. were developed at speeds of 5 and 40 miles per hour, respectively, and average stresses at one edge of 23 000 and 14 000 lb. per sq. in. at the two speeds. Comment may be made that except for the stress of 23 000 lb. per sq. in. the stresses on the 7° curve were relatively low. In the outer rail of the 10° curve of the Lehigh Valley Railroad the highest average vertical bending stress under a driver of the Santa Fé type locomotive was 11 200 lb. per sq. in. at 35 miles per hour. In the inner rail an average vertical bending stress of 21 500 lb. per sq. in. and

an average stress at one edge of 31 500 lb. per sq. in. were found at 5 miles per hour. At 35 miles per hour the corresponding stresses under this driver were 11 000 and 12 000 lb. per sq. in. It should be borne in mind that this inner rail was badly worn, having been relaid from the outer rail.

The cars with wheel loads of 18 000 lb. developed stresses in both rails of the 7° curve of the Baltimore and Ohio Railroad that were lower than the stresses found under the drivers, both vertical bending stresses and stresses at one edge. Similarly, on the 10° curve of the Lehigh Valley Railroad the wheel load of 24 000 lb. developed stresses that were considerably smaller than those found under the drivers of the Santa Fé type locomotive.

It is to be expected that the smaller lateral bending stress in the heavy rail, as well as the smaller vertical bending stress, will give a more general distribution of the lateral and vertical pressure on the ties and ballast, and thus act to decrease the maintenance work required.

34.—Other Tests.—Of the other investigations not reported here, further progress has been made in tests of rail joints. Measurements of strains in angle-bars under the load of wheels have been made on track. Laboratory tests on several of the forms of rail joint commonly used in track have been made from time to time as opportunity afforded, and thus data on the action of rail joints are being accumulated. Tests have been made to find the longitudinal pressure required to cause the rail to slip in the joint for different conditions, such as varied tension in the bolts. It is planned to carry further the investigation on rail joints. It is hoped that an instrument may be devised to measure strains in the splice-bars at the top, bottom, and middle of the bar under the conditions of support of rail found in track. Tests in other lines are also under consideration.

Respectfully submitted,

Special Committee to Report on Stresses in Railroad Track,

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THE INFLUENCE OF THE AUTOMOBILE ON REGIONAL TRANSPORTATION PLANNING*

By George A. Damon,† Esq.

The relation of the automobile to the transportation arrangement within Metropolitan Districts is one which should be discussed by all classes of technical men. It has been said that the difference between the civil engineer and the mechanical engineer is that the civil engineer designs structures to stand still, while the mechanical engineer builds machines to move. Transit facilities need both kinds of talent, to locate and construct the permanent way and to furnish the equipment of motive power and moving vehicles. All classes of engineers are interested in the fundamental principles which underlie the most economical distribution of transportation units to secure the safest and most adequate systems for the movement of both passengers and freight.

Regional Planning.—The art of city planning is rapidly expanding to include the design of great Metropolitan Districts under the title of "Regional Planning." Conferences and commissions are discussing the "interdependence" in matters of common concern of the contiguous communities comprising the areas surrounding great cities. Of all these common welfare problems the question of intercommunication by means of an adequate, coordinated, and efficient system of transportation, is one of the most vital. The development and rapidly increasing use of the automobile now makes available an entirely new solution of the problem of securing the most desirable living conditions for human beings. The traffic congestion on highways and on city streets, due to the automobile, points the way to a new conception of city planning.

Expansion and Contraction.—All engineers know that every action has its corresponding and equal reaction. Thus, in regional development, the conditions at any time are the resultant of the forces of contraction (or congestion) and those of expansion (or decentralization). These opposing forces must be equal in order to maintain an equilibrium or a stable balance. Here, then, is the cue for design. Cities are bound to crystallize about centers where there will be great traffic congestion and hence they must be planned so as to make the most of the urban disadvantages.

To offset this congestion at the metropolitan center, outlying districts must provide for the freest circulation of traffic so as to secure for at least 50% of the population the benefits of suburban or even country surroundings. In working out the balance between city centers, satellite sub-centers, and

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open country, the automobile proves to be the vehicle of expansion while all other transportation units are really most efficient when working under conditions of comparatively higher density of traffic.

Congestion Regulated but not Cured.—Congestion at the centers is inherent in city development and "down town" streets become crowded almost to stagnation with automobile traffic. To widen these streets or open new ones, as they should and will be, is only to invite more automobiles, more traffic, and more congestion.

Building subways takes the cars off the street, and brings more automobiles in the resulting street space. Even the proposed "improvement" of upper decks for pedestrians, will only be temporary and there will soon be more congestion. Every positive solution which involves a physical improvement results in greater congestion, or, in other words, "the cure is worse than the disease." However, engineers will continue to effect these "cures" as fast as economic pressure produces the urge and the where-with-all, but they cannot hope ever to do away with congestion at city centers by any known means of actual physical betterment. As fast as voids are produced at the centers, the forces of centralization will fill them to the saturation limit.

The problem is to use these city centers most efficiently by eliminating the less effective use of the available open spaces. For instance, the parking of private automobiles along the curbs interferes with the movement of traffic and thus, first by limitation and then by complete prevention of "parking", the streets are "regulated" to their higher use. Formerly, people catered to the business of the man in the automobile; now they find that the private automobile with its average load of about 1½ passengers per car is using too much of the valuable space in down town streets and they are going to make it so uncomfortable for the owner that he will leave his machine home and come down town on the motor coach, the street car, or the subway. Therefore, the new conception of city planning, which comes with the continued growth of the use of the automobile, is that down town streets are primarily for the use of pedestrians and that passengers through city centers must be handled *en masse*. Much of the business that depends on the private automobile for service or patronage must move to some outlying sub-center where curb parking is still possible. Ordinarily, this means that when business frontage becomes worth $1 000 per front foot it is subject to traffic restrictions, which will be more and more in favor of the pedestrian and of through traffic and less and less in favor of parking and of the individual in the private automobile.

Decentralization.—Big business has already sensed this movement toward decentralization; branch banks have sprung up on nearly every corner of local business centers, and automobile agencies, accessory supply stores, and oil stations at nearly every "four corners" make convenient shopping centers for the automobile owner. The same automobile which has congested down town streets now can take the owner to a theatre, to a store, to a library, to a dentist, or to some similar place of business or pleasure located in a sub-center.

Thus, the automobile, which is the cause of modern congestion, furnishes the means of escape. The thesis for the design of great Metropolitan Districts, then, becomes "a city center surrounded by satellite sub-centers all interconnected with the best possible transit arrangements."

These sub-centers may even start as an automobile "cross roads" and gradually evolve into satellite sub-centers of no mean importance, but their history is sure to reflect the development of constantly improving transportation facilities. Growth and prosperity are functions of easy access and unrestricted movement. This leads to a consideration of the relation of the automobile to other forms of transportation.

Each Vehicle Has a Place.—In the scheme of transit arrangements each mode of transportation has a place of its own, depending on the density of traffic to be handled, with just sufficient overlapping to make an easy transition from one system to another. Under a given set of conditions, it is possible to prepare curves showing the cost of a passenger-mile, a seat-mile, or a car-mile, which curves will indicate the influence of density on the total cost of operation, including depreciation and interest on the investment. With a private automobile this cost will range from, say, 20 cents to 5 cents per car-mile, depending on conditions of use.

Next in the scale will come the auto-bus or the "motor coach", as people are beginning to call it, with a cost per coach-mile of from 15 to 30 cents, but with the possibility of giving service at the rate of 2 cents per passenger-mile. The trackless trolley is intermediate between the motor coach and the trolley car, which latter vehicle, under favorable conditions, can carry passengers for as low as 1 cent per passenger-mile.

The "rapid transit" car on its elevated structure, in an open cut or in a subway, has the great advantage of capacity as it can run in trains of as many as ten cars, but at no less cost per passenger than the trolley car. For each density of traffic there is a corresponding vehicle for most economical operation so that the transportation system for any given district should be co-ordinated to obtain the best results.

The Place for the Motor Coach.—The private automobile, of course, has no regular place as a commercial carrier in the general transportation plan but the motor coach has a very definite field, which is defined by the first cost and the total cost of operation.

The outstanding advantage of the motor coach is the fact that its gross earnings per annum may be equal to the original investment. A motor coach costing \$10 000 (including repair facilities) can earn \$10 000 per year. Ordinarily, a trolley system can earn in one year an amount equal to only about 25% of its total investment and a rapid transit system often has a gross yearly income of less than 12½% of its first cost. Furthermore, the motor-coach system can be added to, unit by unit, as the business grows and without the delay usually experienced in locating and building tracks and designing and constructing power plants, sub-stations, and distributing systems. The motor coach is a vehicle, then, which can be used on the skirmish line of new business. Wherever there is \$10 000 gross earnings per year per mile in

sight, a bus line can be established giving, say, 30-min. service, and as soon as patronage increases more busses can be added and the headway reduced. One of the important functions of the motor coach, therefore, is to act as a feeder, extending the transit network into otherwise unserved territory.

In the smaller cities motor coaches can run on parallel streets only ¼ mile apart, whereas with a trolley-car system the lines can be maintained only at ½-mile intervals. This development means that in the design of districts it is extremely important to provide through streets every ¼ mile throughout the territory. A population density averaging 5 persons to the acre is potential motor-coach territory, so that the bus may follow the sub-divider very closely.

The Limitations of the Motor Coach.—The very advantage which the motor coach possesses in having each unit carry its own overhead charges works as a disadvantage when density of traffic develops. With this system additional units do not reduce the overhead charges on each unit as with a trolley system, where each car must stand its proportion of the costs of the tracks, feeder system, and power plants. The result is that, when good service demands a vehicle every 10 min., or oftener, the motor coach must give way to the trolley car. The cost of carrying a passenger per mile cannot be reduced to much less than 2 cents in a motor coach, whereas a trolley-car system can reduce this cost to as low as 1 cent per passenger-mile, if a certain percentage of the patrons are allowed to stand during the rush hours. At about a 10-min. headway, the two systems overlap and, at this point, the pioneer motor-coach system may well turn over its business to the trolley-car system, and seek other territory to explore.

"De Luxe" Transportation.—The modern motor coach can now take the place of the private automobile on the down town streets, and along the boulevards, to supply transportation at a charge of approximately twice the street-car fare. This kind of service should not be considered as competitive with street cars as it occupies a field which the trolley car does not cover. The obvious place for the electric street car is within a radius of about 5 miles from the center of the city, running through from one side to the other—with one fare (preferably 5 cents) and universal transfers. The motor coach can handle the patronage beyond the 5-mile limit for a fare of 10 cents and higher for a ride directly to the city center, and such service would be an acceptable substitute for the private automobile which is now handicapped with so many hardships when it reaches the business center.

Street-car systems have been operating at a gross annual income of from \$15 to \$25 per capita. Considering that many families of four maintain an automobile at a cost of more than \$100 per month, or at the rate of \$300 per person per annum, at once, there appears the possibility of raising the earning power of the motor-coach system by encouraging the riding habit with better and better service and equipment.

Other Applications of the Motor Coach.—There are many other uses for the motor coach in building up the community. Wherever there are a dozen people desiring transportation to any given place at the same time there will be a use for the public automobile.

Not only may the motor coach be used as a broadcasting agency for the trolley system or for the rapid transit lines, but it can actually be used to parallel existing street-car lines as an auxiliary during rush hours.

Sometimes an isolated satellite community will subsidize busses to give either free rides or reduced fares on a bus line connecting with a stop on the near-by trolley of the interurban line.

District high schools are using motor coaches to collect and distribute their pupils over a large district, thus making higher education available to many families living in the country.

Auto stages are making regular trips over constantly improving roads throughout the country and these independent stage lines are being consolidated into unified systems as rapidly as the advantages of co-ordination are realized. The existing steam and electric railroads are beginning to recognize the value of the automobile, not only as an auxiliary, but as an integral part of a well-balanced system of transportation service to the entire district. This comparatively new vehicle for public transportation promises to carry its fair share of the great increase in passenger and freight movement which grows much faster than the population.

Zoning and Automobiles.—Just as the private automobile enabled many well-to-do people to live in the country and yet enjoy most of the advantages of city life, so the motor coach is now making it possible for all classes of people to own a home in a district zoned for single family residences.

Street-car systems and multi-family residence districts seem to go hand in hand and local commercial centers spring up naturally along the car tracks. Rapid transit lines mean contiguous districts for apartments and business centers filled with skyscrapers. Tracks mean more or less of congestion for a built-up city, whereas the motor coach, with its ability to earn its way and with comparatively a much lighter gross income, can be made the means of escape from the disadvantages of congestion. "Rails", therefore, are the indication of a tendency toward congestion or contraction, while "rubber" is the indication of the opposite tendency toward the open country or expansion. Both mvements need encouragement, and in all zoning regulations, the relation between the public automobile and other forms of transportation should be recognized.

Regional Planning Specifications.—As an example of the application of the fundamental principles of regional planning as affected by the continually increasing use of the automobile, the speaker submits the following requirements which have been adopted and are now being used in the practice of the Los Angeles County Regional Planning Commission.

Based on the highway map in existence as of January, 1923, a regional map has been prepared showing a complete network of "through" roads, which are to provide automobile traffic circulation to every part of the region. This plan provides major highways, not less than 90 ft. (and preferably 100 ft.) in width between property lines about 1 mile apart—often on the section lines in the valley districts.

On the half-section lines or at about halfway between the major highways are laid out intermediate highways 80 ft. wide. A right of way of this width

will allow a pavement of 56 ft. between curbs, which is just sufficient for four lines of moving vehicles with a parking space along each curb.

On the quarter-section lines, or at a distance of approximately ¼ mile from the major highways and the intermediate highways are rights of way 60 ft. wide for the minor through highways. These streets will provide for pavements 36 ft. in width, allowing for two lines of moving vehicles plus the usual parking space along the curb.

All other streets that can possibly be termed "through streets" are made not less than 60 ft. in width, and only where they are of a strictly local character or in hillside sub-divisions are streets of less than 60 ft. in width approved.

An interesting detail is the requirement of rounded corners which provides for a radius of not less than 30 ft. at the curb line at the intersections of all streets; this arrangement contributes considerably to the capacity and safety of the street.

This highway plan contemplates eventual motor-coach service at least every ¼ mile throughout the district. This will naturally serve satellite business centers, scattered through the region at convenient and economical strategic positions. At these sub-centers there should be convenient transfer facilities to the electric rapid transit or to the suburban lines, connecting directly to the main metropolitan business center.

At a short distance from the rapid transit line, or sometimes directly contiguous to it, is a double highway, with a roadway on each side of the track, connecting each sub-center with its neighboring center. These parallel highways provide a right of way for a "local" motor-coach service carrying passengers between the "express" stops on the rapid transit electric lines. This arrangement makes it possible to cut out the local stops on the electric lines and devote the rails to express service with stations about 2 miles apart resulting in the possibility of reaching the periphery of a 20-mile radius by means of "rapid transit" in less than 1 hour from the time of leaving the central station.

Business Sub-Centers.—Actual development of sub-centers shows that, as a rule, each grows up about a trading center, which crystallizes around a nucleus at some intersection of the major highway system. Therefore, each "four corners" is a potential sub-center and the subdivider very naturally lays out his plot at these intersections as business property. Business lots are also scattered along the major highways (sometimes the entire length of the highway), thus providing many times more business frontage than will be actually needed for years to come.

In Los Angeles County, wherever business is contemplated, the street is to be 100 ft. wide to allow for 70 ft. of pavement and 15 ft. of sidewalk space. Each business block is to have an alley not less than 20 ft. wide to allow for the delivery of parcels and goods from the rear of the stores.

Corner lots are preferably to be of double width to allow for a "cut-off" corner at the street intersection. A radius of 15 ft. at the property line or a diagonal cut-off connecting set-back points 12 ft. from the corner increases the sidewalk space and the visibility sufficiently to justify this requirement.

A Set-Back Line Proposal.—One of the most important highway developments that has been evolved from regional planning activities is the conviction that there must be "set-back" lines to provide for future requirements.

The use of the automobile required streets not less than 100 ft. in width at all business centers. This width will allow for a through traffic strip for four lines of vehicles at the center, and two widths of 15 ft. each along the curbs for local parking; 70 ft. of pavement and 15 ft. of sidewalks is the irreducible minimum for a business street anywhere in the Los Angeles District. Probably the quickest way to realize this requirement is to adopt a regulation that "no structure shall be built nearer than 50 ft. to the center of any through street."

If the automobile is to occupy its proper place in the community as a common carrier, if the convenience of the sub-center is to be encouraged, and if there is any hope of controlling the forces of decentralization which are at hand to offset the disadvantages and even evils of city congestion, the police powers at hand must be wielded to provide for adequate open spaces for the circulation of traffic.

Grade Elimination.—Of equal importance to capacity are the requirements for additional safety. Railroad trunk lines and electric rapid transit rights of way must be separated from the grade of the automobile highways. In the ultimate analysis this means the unification and, very probably, the electrification of transcontinental railroad terminals. The expenditures called for by these improvements are staggering in their totals and can be justified only by the fact that tonnage increases as the square and sometimes as the cube of the population. This growth and anticipated expenditures call for a comprehensive plan and a logical, consistent, step-by-step program. With such a plan and program in mind, the separation of grades on main highways may be begun at once, making each expenditure part of the ultimate result.

It is hard to visualize the conditions which will exist on highways when there are ten times as many automobiles as at present, but it is certain that the automobile traffic will be increased by ten times at no far distant date. "More roads, wider roads, and safer roads" must be the slogan in this struggle with traffic and congestion; until there is complete grade separation the highways will not be entirely safe.

One of the next big adventures in regional planning will be the construction of safety highways located, perhaps, on rights of way over and through congested districts. Bridges over rivers and railroad tracks in the industrial districts will naturally be elongated to provide safe passage over switch tracks, and such viaducts will point the way to the more extensive use of this kind of an "elevated" for the use of the automobile.

Recapitulation.—The Mobility of the Automobile.—The flexibility of the automobile is the controlling feature which makes it available as an addition to present transit facilities of equal importance with the steam locomotive and the electric car as has already been emphasized.

It can be installed on the "unit" basis and each unit carries within itself its own power plant and its own overhead charges.

The first investment per seat in a "motor coach" to be used as a common carrier is about one-fourth of the investment required by a trolley-car system so that it can be used to advantage as a pioneer for new business in comparatively sparsely settled districts.

It is the only vehicle which does not depend on density of population for its support, in fact as soon as congestion appears the motor coach is ready to turn over its load to be carried on "rails" instead of "rubber". The automobile, therefore, is the exponent of expansion.

Decentralization is the natural result of the congestion caused by the use of the automobile and the same vehicle which causes the congestion of street traffic, furnishes a means of escape, provided enough roads are built to carry it around and away from the city centers.

The very danger which the automobile has brought at grade crossings and on the highways will result in the building of "off-grade" structures which will provide safety highways for the future and which would have been financially impracticable with less density of traffic.

Regional planning itself is the direct result of the use of the automobile which has made time and not distance the limit of modern activities and influence. Any locality within a distance of an hour's automobile ride is now an integral part of the Metropolitan Area and its physical improvement is a matter of common concern to every inhabitant in the district. Safety, accessibility, convenience, speed, economy, adequacy, and co-ordination are requirements alike for the steam roads, the electric lines, and the motor-coach systems which provide "mass transportation" to all the people of the great residential districts.

Prosperity is a function of activity. The economic urge toward increased transit facilities has been greatly emphasized by the use of the automobile. These vehicles have been produced much faster than the roads on which they operate. In furnishing transportation service for future generations, or even for the immediate future, "rails" must be loaded to their full carrying capacity and "rubber" must be provided with a network of through roads, all interconnected to avoid congestion in the outlying districts and to provide for the orderly direction of traffic at the city centers.

SECONDARY STRESSES IN BRIDGES

Discussion*

By Messrs. Edward Godfrey and Cyrus C. Fishburn.

Edward Godfrey,† M. Am. Soc. C. E. (by letter).‡—In discussing the subject of secondary stresses of course the first thing to establish is a definition of the term. There is a sharp difference of opinion as to the importance of secondary stresses, and little can be accomplished in analyzing this difference of opinion unless the term "secondary stress" is first defined and understood.

Engineering terms must be defined by engineers. The dictionary is of little or no use except to give the general meanings of the words. The engineering application of these words is a matter for engineers to agree upon or to develop by custom and usage, guided somewhat by authoritative utterances. Many engineering terms are not found in the dictionary, though they are in common use among engineers and have definite meaning.

In general, primary stresses are well understood. They include axial tension or compression in members ordinarily designed for such stresses. They include bending in any member subject to transverse loading, whether this loading be an extraneous load or the weight of the member itself. In other words, a compression or tension member, as the chord of a truss which carries floor ties, is not subject to secondary stress by reason of this transverse loading, for both the direct axial load and the transverse load are of primary importance in the design of these members. In the same manner, a long horizontal member must be designed to carry its own weight, as a primary consideration, and it is only when custom and experience have proven that members are so short as to give bending stresses, due to their own weight, of no consequence in the design, that these stresses may be neglected.

Stresses that are produced by wind loads, whether direct axial stresses in truss members or bending stresses created in these members by wind shears that must be carried by the truss members to an adequate resisting medium, are not secondary. They are primary, and when of considerable magnitude must be accounted for in the design. Only when they are small in comparison with the main truss stresses or floor system stresses is there any warrant for neglecting them.

Stresses that are produced by an eccentric application of the direct load of a tension or compression member are not secondary. Specifications require

* Discussion on the paper by Cecil Vivian von Abo, Jun. Am. Soc. C. E., continued from March, 1925, *Proceedings*.

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‡ Received by the Secretary, November 3, 1924.

that axial stresses be applied at the center of gravity of the cross-section of a member and imply or express the requirement that if axial stresses are not so applied, the bending moments produced must be provided for.

When a girder or beam rests on a bracket on a column without web connection and without connection of both flanges to the column, there is a definitely calculable bending moment in the column due to this bracket load, and the column must take all this bending moment—the girder or beam cannot take any part of it. The bending moment is not a secondary stress in the column. It is conceivable that such a load could wreck the column, the stress being greater in amount than the direct axial load on the column.

In all the foregoing cases there can be no dispute, in proper designing, as to whether and when the stresses referred to should be accounted as worthy of consideration. When by ordinary calculation these additional stresses amount to a considerable fraction of the axial stresses, they must be governing factors in proportioning the members. Safety in structures demands this.

Some of the cases that will doubtless be conceded by all to be classed as secondary stresses are the following:

(*a*) In a column to which a girder is rigidly attached—the bending stresses due to the deflection of the girder. It is assumed that the plane of the web of the girder coincides with the axis of the column.

(*b*) In a truss rigidly riveted at all joints—the bending stresses due to the deflection of the truss.

(*c*) In a pin-connected truss—the bending stresses due to the tendency of the members to rotate as the truss deflects, the friction on the pins supplying the force.

All these have the common charactertistic that they are almost universally ignored by designers.

In designing, a difference of opinion as to the precise use of the words "secondary stresses" is of little importance. As intimated in the beginning of this discussion, however, the sharp difference of opinion as to whether secondary stresses are worthy of consideration in design is of great importance and the line of demarcation in this difference of opinion is that outlined in the foregoing classification. It is pertinent, in considering a subject of this nature, to inquire into the basic assumptions and to establish their soundness. If extremely complex mathematics must be used in the design, it is particularly appropriate to establish the necessity for that method of design; otherwise engineering is burdened with unnecessary complications.

One of the simplest structural problems is that of designing, for a given loading, two columns and the girder connected to and supported by them. If this girder rests on brackets on the sides of the columns, with no web connection and no top flange connection, the span is the distance center to center of these brackets, and the bending moment on the column is simple to calculate. This is not good construction, however, and would not ordinarily be considered good engineering, because lacking in rigidity.

Suppose, then, the girder rests across the top of the column or has a riveted web connection to the side of the column or has top and bottom flange con-

nections to the column. Any one of these accepted and proper details of construction creates a condition that, theoretically, demands consideration of bending moments in the columns due to the deflection of the girder. The same condition theoretically would permit the design of the girder for a span less than the distance between the columns. No practical designer (that is, no person actually engaged in the practical design of structures) ever designs columns for these bending moments, nor does a practical designer who is careful ever use less than the distance center to center of the supporting posts for the span of a girder, where the posts support no other load than that transmitted by this girder.

The burden laid on the practical designer would be great if he were required to apply the method of least work in the solution of this simple problem. It would be enormous, in fact, unbearable, if he were compelled to follow this method throughout in the design of a small building.

General acceptance of a system of design has no weight if logic or tests can prove it erroneous, but logic and tests must show a pretty good case before they can displace a generally accepted standard of design.

The best tests are the practical tests of existing structures. The writer believes he can state without fear of contradiction that no steel structures have failed because the designers have neglected to take into account the secondary stresses due to deflection of the girders. He will go a step further and state that he does not believe it is possible for a structure to fail, if the ordinary rules of design are followed, from such cause as the deflection of the girders.

The argument of logic is in favor of the standard method of designing and against the ultra-theoretical. If steel were brittle and perfectly elastic, like glass, this would not be the case—all calculable stresses would have to be accounted for; but steel has a property that makes it safe for structures, the property of toughness or malleability. Theory can no more ignore this property than it can ignore any other known property of a structural material.

Consider the following two cases where the elastic theory shows exactly equal stress in the columns. Each case consists of a pair of columns and a girder supported by them. All four columns are identical in cross-section; but one girder is connected rigidly to the columns, and the other simply rests on brackets, the distance from center of column to bracket support being 3 ft.

The girders are ample in section for their loads, but the columns are weak to the extent that the unit stress, computed theoretically, is the ultimate for the columns, although the columns are capable of taking the direct stress due to the girder load. The theory by which the unit stress in the rigidly connected system is computed is that which considers relative rigidities of the columns and girders and utilizes the method of least work or some other involved process, and finds imaginary points of inflection 3 ft. from the centers of the columns. Thus the unit stresses, theoretically, are identical when the systems are loaded.

The system in which the girder rests on the brackets will fail. The other system will not fail.

The first system will fail because there is nothing whatever in it to come to the aid of the columns when they deflect in the act of failing. The second system will not fail because the girder prevents the columns from deflecting to the point where they will crush or buckle.

These facts can be reasoned out by analysis, starting with the known properties of steel. They can be demonstrated by test. Every engineer who has examined many existing bridges is familiar with the fact that there are cases without number where the theoretical stresses, if realized, would wreck the structures in which they occur. These facts provide a sound basis for the general disregard of secondary stresses as defined in the present discussion. Unfortunately some designers do not discriminate between computed stresses that may with impunity be ignored, and that are among those classed as secondary in this discussion, and the other stresses which are herein defined as not secondary.

What is said of the effect on a structure of secondary stresses due to the deflection of a girder is equally true of secondary stresses in truss members due to the deflection of a riveted truss or a pin-connected truss, either because of the rigid joints or because of rotation on the pins. The point of distinction between the case that needs consideration and the one that does not is the possibility of continued deflection. If a member can deflect only so far and is positively prevented from deflecting any farther, and if reverse deflection (that is, bending back and forth) is not a possibility, it is known from the properties of steel that, in general, failure is impossible.

A bar, rod, or wire can be bent over a corner or around a curve into such a shape that calculations, based on the perfect elasticity of steel, would show ultimate stresses in the piece, and yet in this shape it will carry a load practically if not fully equal to its capacity as a straight member. Has theory any right or warrant to say this member is unsafe, when countless members in this identical position are doing full service without any distress whatever? This bent member will carry the load imposed on it indefinitely—forever, in fact—barring deterioration. What standing has the theory which declares the member to be dangerously stressed?

On the other hand, if the piece were bent back and forth repeatedly, even through a much smaller angle than would be safe for the first bend, it would soon, or eventually, break. This behavior, as well as the other, can be reasoned from the known properties of steel—its toughness and malleability; and these properties are just as real and, when understood, just as dependable as the elasticity of steel, which is the only recognized property of the metal that the average theorist uses in his theorizing.

There are grades of steel, very high in carbon and very brittle, that would not stand safely even one bend that other steels would stand with perfect safety; but structures are not made of the hard, brittle grades of steel. It is not proper to introduce into the theory of structures the properties of a steel that is not used in the fabrication of structures. Economic, safe, and proper design is not advanced thereby.

Consider the two pairs of posts in Fig. 111. The posts of one pair, *A*, have been jacked apart and a ball placed between them. The posts of the other

pair, *B*, are bowed in their fabrication. All posts are of the same cross-section. The unit stress from direct load in each of the four posts is the same, namely, 10 000 lb. per sq. in. The unit stress from bending is, theoretically, exactly the same, namely, 12 000 lb.; but the bending moment in one pair is due to the wedging or springing action of the ball forced between the columns, while that in the other pair is due to the bowed shape of the columns and the eccentric application of the load.

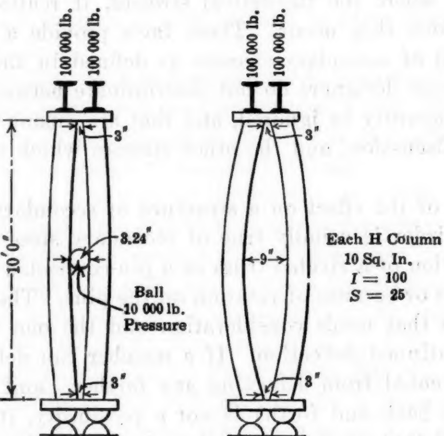

FIG. 111.

In spite of the identical stresses in these two pairs of columns, theoretically determined, they are by no means equally unsafe. The writer would not hesitate to condemn the pair of columns shown at *B*, if he found such a condition in a structure. He would not be greatly concerned for the safety of the structure, if he found the condition at *A* existing.

The condition at *B* is a menace to the integrity of the columns, and failure, once started, would continue without hindrance. Part of the deflection of 3 in. (about ⅛ in.) is produced by the load which the columns are supporting. There is, therefore, the start of failure in the columns as shown. The load will not cause any deflection in the columns at *A*, but will relieve some of the pressure on the ball.

In the fabrication of "fish-belly" compression members the parts are bent cold. This puts into the metal initial stresses that correspond to, in fact, are much greater than, those found by the theorist in members of a truss as due to secondary stress. The lattice-bars in these members with bowed parts, however, hold these parts against deflection, and no weakness is developed in the members. In exactly the same manner the theoretical unit stresses found in the members of a truss due to the secondary stress caused by deflection of the truss mean nothing, because their effect ceases at their inception and cannot be augmented. In fact, it is very doubtful if they actually exist. The tests on the Hell Gate Bridge showed far from consistent and expected results.

Malleable metal, such as structural steel, can take up and absorb such stresses. They mean little or nothing as regards safety.

D. B. Steinman, M. Am. Soc. C. E., who made the tests referred to, states:* "For the highest percentages of secondary stresses, the measured values are only a small fraction of the calculated values". Again, "The actual secondary stresses will generally be lower than the calculated values. There is an automatic readjustment of strains within a structure in such direction as to relieve the secondary stresses".

Referring to measurements made on buildings it has been stated† that: "At certain points of heavy columns where one might reasonably expect considerable bending to be developed by distortion or secondary effect, there is generally speaking no bending". As a result of tests of secondary stresses, O. H. Ammann, M. Am. Soc. C. E., states:‡ "Eccentric connections increase the secondary stresses almost invariably and should therefore be avoided". Such stresses are not true secondary stresses. He also states, "Rigid riveted connections, on the other hand, always act favorably, distributing the moments and reducing the stresses from eccentric connections". It is rigid connections which, theoretically, cause the greatest secondary stresses. He states again: "The results of the investigation indicate that rigidity of members and connections tends to prevent localization of high secondary stresses and may therefore constitute an element of safety rather than of weakness".

Imperfections in workmanship can be shown theoretically to produce far greater stresses in structural members than are produced by deflections of these members in the structure. Suppose, for example, that a girder is milled out of true a small fraction of an inch. When this girder is riveted to the column, the column will be bent to a far greater extent than the deflection of the girder would bend it, yet a practical engineer, understanding the properties of steel, would not concern himself in the least if this condition were found to exist in his structure.

Again, suppose a column or truss chord were milled out of true. The calculated stress could be enormous in intensity—sufficient to break any material that is brittle like glass or concrete. In steel the member may show no distress whatever. Of course, careless workmanship is not a thing to be desired or encouraged, but any engineer who, as the writer, has examined in detail a hundred or more existing structures and observed these imperfections, and has seen metal peened down by reason of its malleability (under high stress), by stresses that, theoretically, would wreck a structure, is aware that malleability is a property of steel that can be relied upon under certain conditions.

The outstanding example of a structure that has many varieties of secondary stresses due to its design—stresses of theoretically great intensity—is the Firth of Forth Bridge in Scotland, but none the less this bridge is relatively one of the safest in existence, because of its design and because of the very rigidity of the joints that gives rise to the theoretical secondary stresses.

* *Transactions*, Am. Soc. C. E., Vol. LXXXII (1918), p. 1071.

† *Engineering News-Record*, October 23, 1924, p. 655.

‡ *Loc cit.*, p. 668.

If the theory of secondary stresses and the perfect elasticity of steel is worthy of consideration in design, there are other cases where it would have application. One of these is a simple web splice in a girder. The writer knows of an eminent engineer, now deceased, who insisted that every web splice should be proportioned for bending or flange stress as well as shear, for the reason that the bottom edge of the web plate elongates and the top edge shortens as the girder deflects. As designers know, it is customary to splice the web plate for its shear only, unless it is desired to take advantage of the aid a web plate supplies in taking up flange stress. In many girders, particularly light ones, there is no need of including the bending value of the web, and a single row of rivets on each side of the cut is sufficient for the shear. To splice the web plate for bending would require four or more vertical rows of rivets, or extra horizontal plates—an expensive detail merely to satisfy a theory; for consider what could happen if a web plate is spliced for shear only, say, with a single row of rivets on each side of the cut. Suppose the extreme rivets should slip an insignificant trifle (the stretch of the flange angle in 3 in.); this would not impair their value for taking shear one iota. With the flanges proportioned for the full flange stress, there is absolutely no weakening of any kind in the girder.

In a triple riveted splice in a boiler or tank seam (Fig. 112), if the elastic theory be applied in a manner similar to the way it is applied in computing secondary stresses, it will be seen that there cannot be equal distribution of stress among the three rivet lines. In one plate the intensity of stress is twice as great as in the other plate, where the two plates lie side by side. Ignoring malleability, this joint would not be safe, for there is no way to work out equal shear on the three rows of rivets and simultaneously equal stress on the contiguous layers of plate. Quadruple riveted joints and, in fact, almost all splices offer greater theoretical difficulties than these.

As the whole fabric of the secondary stress theory rests on the soundness of its fundamental assumptions, and as it is proven by the cases presented that these fundamental assumptions are without warrant in fact, or experience, or test, it is clear that there is no justification for consideration of secondary stresses nor for adding the complexities of that theory to the literature of engineering.

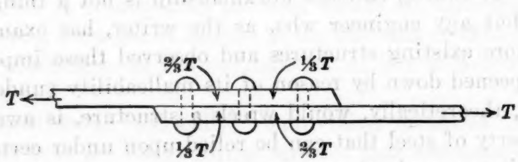

Fig. 112.

In the last few years two bascule bridges have exhibited failure in the members that supported the heavy counterweights. Statements have been published to the effect that lack of consideration of secondary stresses in the design of these counterweight supports was responsible for the failures, and engineers who advocate the ignoring of secondary stresses are held accountable for these failures and any others of a similar character that may occur.

Little is given in the way of details of the supports of these counterweights. However, one thing is made clear, namely, that the counterweights are of concrete cast around built-up truss members. The raising and lowering of the counterweight causes these structural members to bend back and forth as one would bend a wire to break it. It is not to be expected that truss members would withstand this bending back and forth due to a heavy counterweight moving through a large angle in changing its position. If loaded bridges were to be turned upside down and all stresses and moments reversed many times a day, this feature would have to be considered in their design, just as a revolving axle must be designed with a larger factor of safety than is necessary for a stationary member in bending.

The writer has yet to find any case where secondary stress is worthy of computation and consideration in design.

CYRUS C. FISHBURN,* ESQ. (by letter).†—In using Mohr's semi-graphical solution for the moments at the ends of the members of a truss due to the change in length of the members, the following check is useful in verifying the equations before solving for the unknowns, thus insuring against arithmetical errors in the original equations.

The Mohr semi-graphical method first requires the deflections of each member of the truss, which are obtained by means of the Williot diagram. An equation is then written for each joint of the truss, the unknowns being the rotation or angular change at the joints. These equations are derived from the basic equation:

$$M_{AB} = 2\,E\,K_{AB}\,(2\,\theta_A + \theta_B - 3\,R)$$

in which, M_{AB} is the moment at the end, A, of the member, $A\,B$, K_{AB} is equal to $\frac{I}{L}$ (movement of inertia divided by length) for the member, $A\,B$, θ_A and θ_B are the rotations at the ends, A and B, respectively, and R is equal to $\frac{d}{L}$, d being the deflection of $A\,B$. From this fundamental equation, a series of other equations is derived by equating the sum of all the moments at any joint to zero. The equation for Joint A (Fig. 113) is:

$$\begin{aligned}&[2\,\theta_A\,K_{AB} + \theta_B\,K_{AB} - 3\,(R\,K)_{AB}] + [2\,\theta_A\,K_{AC} + \theta_C\,K_{AC} - 3\,(R\,K)_{AC}]\\ &+ [2\,\theta_A\,K_{AD} + \theta_D\,K_{AD} - 3\,R\,K)_{AD}] + [2\,\theta_A\,K_{AE} + \theta_E\,K_{AE} - 3\,(R\,K)_{AE}]\\ &= 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)\end{aligned}$$

which may be written in the form:

$$\begin{aligned}&2\,\theta_A\,(K_{AB} + K_{AC} + K_{AD} + K_{AE}) + \theta_B\,K_{AB} + \theta_C\,K_{AC} + \theta_D\,K_{AD}\\ &+ \theta_E\,K_{AE} = 3\,[(R\,K)_{AB} + (R\,K)_{AC} + (R\,K)_{AD} + (R\,K)_{AE}] \ldots\ldots (2)\end{aligned}$$

It is to be noted that the left-hand member of Equation (2) contains the rotation, θ, of both ends of all members entering the joint, having as coefficients the K's of the members. The right-hand member of Equation (2) contains $3\,R\,K$ for all members.

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† Received by the Secretary, February 13, 1925.

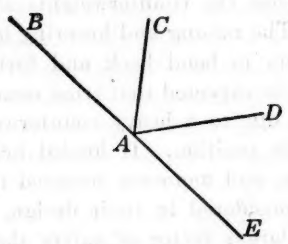

FIG. 113.

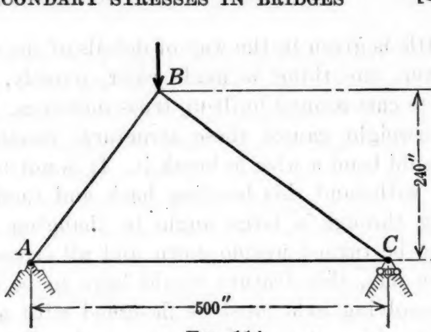

FIG. 114.

The equations for all joints in the structure of Fig. 114 are presented in tabular form in Table 36.

TABLE 36.—FUNCTIONS OF JOINTS.

	Equation.	$\theta_{A'}$	$\theta_{B'}$	$\theta_{C'}$	Right-hand member of equation.
Joint A	(1)	$2(K_{AB} + K_{AC})$	K_{AB}	K_{AC}	$= 3\,[(RK)_{AB} + (RK)_{AC}]$
Joint B	(2)	K_{AB}	$2\,(K_{AB} + K_{BC})$	K_{BC}	$= 3\,[(RK)_{AB} + (RK)_{BC}]$
Joint C	(3)	K_{AC}	K_{BC}	$2\,(K_{AC} + K_{BC})$	$= 3\,[(RK)_{AC} + (RK)_{BC}]$

Before setting up Equations (1) and (2), it is convenient to tabulate the geometrical properties of the members. Such a tabulation for the simple truss in Fig. 114 is presented in Table 37.

TABLE 37.—PROPERTIES OF MEMBERS.

Member. (1)	Length, L, in inches. (2)	Area, A, in square inches. (3)	I, in inches⁴. (4)	$K \frac{I}{L}$. (5)	Stress, P, in pounds. (6)	$\frac{PL}{AE}$, in inches. (7)	Deflection, d, in inches. (8)	$R \frac{d}{L}$. (9)	$3RK$ $3I\frac{d}{L^2}$. (10)
AB	300	32	3 000	10	− 500 000	− 0.1562	+ 0.408	+0.00136	+0.0408
BC	400	24	1 600	4	− 375 000	− 0.208	− 0.307	−0.000768	−0.009216
AC	500	20	1 500	3	+ 300 000	+ 0.250	0	0	0
Totals...	...	..	.....	17	......	....	......	....	+0.03158

It may be seen by inspection of the equations in Table 36 that the sum of the coefficients of θ in all three equations equals $6\,(K_{AB} + K_{AC} + K_{BC})$, or six times the sum of all the K's in Column (5) of Table 37. Similarly, the sum of the right-hand members of these equations equals $6\,[(RK)_{AB} + (RK)_{AC} + (RK)_{BC}]$, or twice the algebraic sum of the quantities in Column (10) of Table 37. This check applies to any truss, irrespective of the number of joints.

The numerical value of the quantities of Table 36, are substituted from Table 37, in Table 38. The sum of the arithmetical coefficients of the θ's in Equations (1), (2) and (3) of Table 38 equals 102, or six times the total, 17, of Column (5), Table 37. Likewise, the algebraic sum of the numerical values of $3\,R\,K$ in these equations, equals twice the algebraic total, 0.03158, of Column (10), Table 37.

TABLE 38.—Numerical Check.

Equation.	θ_A	θ_B	θ_C	Right-hand member of equation.	Check column.
(1)	+ 26	+ 10	+ 3	+ 0.0408	+ 79.8
(2)	+ 10	+ 28	+ 4	+ 0.03158	+ 73.58
(3)	+ 3	+ 4	+ 14	− 0.00922	+ 11.78
(4)	+ 1	+ 0.385	+ 0.115	+ 0.001569	+ 3.069
(5)	+ 1	+ 2.8	+ 0.4	+ 0.003158	+ 7.358
(6)	+ 1	+ 1.333	+ 4.667	+ 0.003073	+ 3.927
(5) − (4) = (7)	.....	+ 2.415	+ 0.285	+ 0.001589	+ 4.289
(5) − (6) = (8)	.....	+ 1.467	− 4.267	+ 0.006231	+ 3.431
(7)	.....	+ 1	+ 0.116	+ 0.000658	+ 1.774
(8)	.....	+ 1	− 2.903	+ 0.00424	+ 2.337
(7) − (8) = (9)	.....	....	+ 3.019	− 0.003582	− 0.563
			$\therefore \theta_C = -0.001187$		

After checking the accuracy of the equations in the manner shown, the work of solving them simultaneously may be continually verified by the well known method of carrying a check column. Such a check column is illustrated in Table 38.

The check column contains the algebraic sum of all the factors in each equation, and it is treated in the same manner as the quantities in the equations. It may be found convenient to multiply the right-hand members of the equations by a constant factor, 100 or 1 000, before adding it to the check column. The factor used in Table 38 was 1 000.

By checking the arithmetical quantities in the original equations, as written and by checking the successive steps in the solution of these equations, the chance of arithmetical error in the result is reduced to a minimum.

DESIGN OF SYMMETRICAL CONCRETE ARCHES

Discussion*

BY MESSRS. R. R. MARTEL, E. H. HARDER, CHARLES W. COMSTOCK, AND A. G. HAYDEN.

R. R. MARTEL,† ASSOC. M. AM. SOC. C. E. (by letter).‡—The use of the influence lines for arches given in the latter part of this thorough and painstaking paper will prove a great aid in determining the preliminary shape of arch axes and the economical cross-sections at the crown and springing lines; but with large spans, especially where the shape of the arch axis and the variation in the cross-sections throughout do not conform closely with the data assumed in obtaining these influence lines, it will still be desirable and in some cases necessary to make analyses of stresses, using the actual data. Opinions regarding the simplest and best method of doing this will be influenced, to a large degree, by the familiarity of the user with the method he sponsors.

To those who are not already committed to one of the older processes of analysis the method presented here may appeal. It has the advantage of avoiding the use of the funicular polygons, being comparatively simple and largely mechanical. Instead of dividing the arch axis so as to make $\frac{ds}{I}$ a constant, or of using arbitrary finite divisions of the arch axis and replacing the integrals in the expressions for crown thrust, moment, and shear, with summation signs, the values of the functions involved are plotted to scale and integration is performed by quadrature; that is, by finding the area under the curves of these functions by planimeter or similar means.

PART I.—ARCH ANALYSIS WITH PLANIMETER

The equations for the crown thrust, shear, and moment, in an elastic arch under vertical loading, given herewith are readily understood and will not be derived here.

$$H_0 = \frac{\int \frac{ds}{I} \int \frac{m\,y\,ds}{I} - \int \frac{m\,ds}{I} \frac{y\,ds}{I}}{2\left[\int \frac{ds}{I} \int \frac{y^2\,ds}{I} - \left(\int \frac{y\,ds}{I}\right)^2\right]}$$

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* Discussion on the paper by Charles S. Whitney, M. Am. Soc. C. E., continued from March, 1925, *Proceedings*.

† Assoc. Prof., Civ. Eng., California Inst. of Technology, Pasadena, Calif.

‡ Received by the Secretary, February 6, 1925.

$$H' = \frac{\Delta \int \frac{d\,s}{I}}{2\left[\int \frac{d\,s}{I} \int \frac{y^2\,d\,s}{I} - \left(\int \frac{y\,d\,s}{I}\right)^2\right]}$$

$$V_0 = \frac{\int \frac{(m_L - m_R)\,x\,d\,s}{I}}{2\int \frac{x^2\,d\,s}{I}}$$

$$M_0 = \frac{\int \frac{m\,d\,s}{I} - 2\,H_0 \int \frac{y\,d\,s}{I}}{2\int \frac{d\,s}{I}}$$

$$M' = \frac{-H' \int \frac{y\,d\,s}{I}}{\int \frac{d\,s}{I}}$$

For a symmetrical arch the limits of all the integrals are for half the arch.

In these equations:

ds = differential length of the arch axis;

I = moment of inertia of any section (I of concrete area + $\frac{E_s}{E_c} \times I$ of steel area);

x, y = co-ordinates of any point on the arch axis with the crown as origin, and all taken as positive;

l = length of span;

E_c = modulus of elasticity of concrete;

E_s = modulus of elasticity of steel;

m = bending moment at any point due to external loads considering each half of the arch as a cantilever. Where m occurs without subscripts, it includes the bending moments at symmetrical points on both halves of the arch. In all cases, m has been taken positive for downward loads;

$\Delta = c\,t\,l\,E_c$ in the case of temperature change;

$= p\,l$ in the case of rib-shortening due to thrust;

$= k\,l\,E_c$ in the case of shrinkage;

c = coefficient of expansion;

t = temperature rise or fall, taken as positive for a rise;

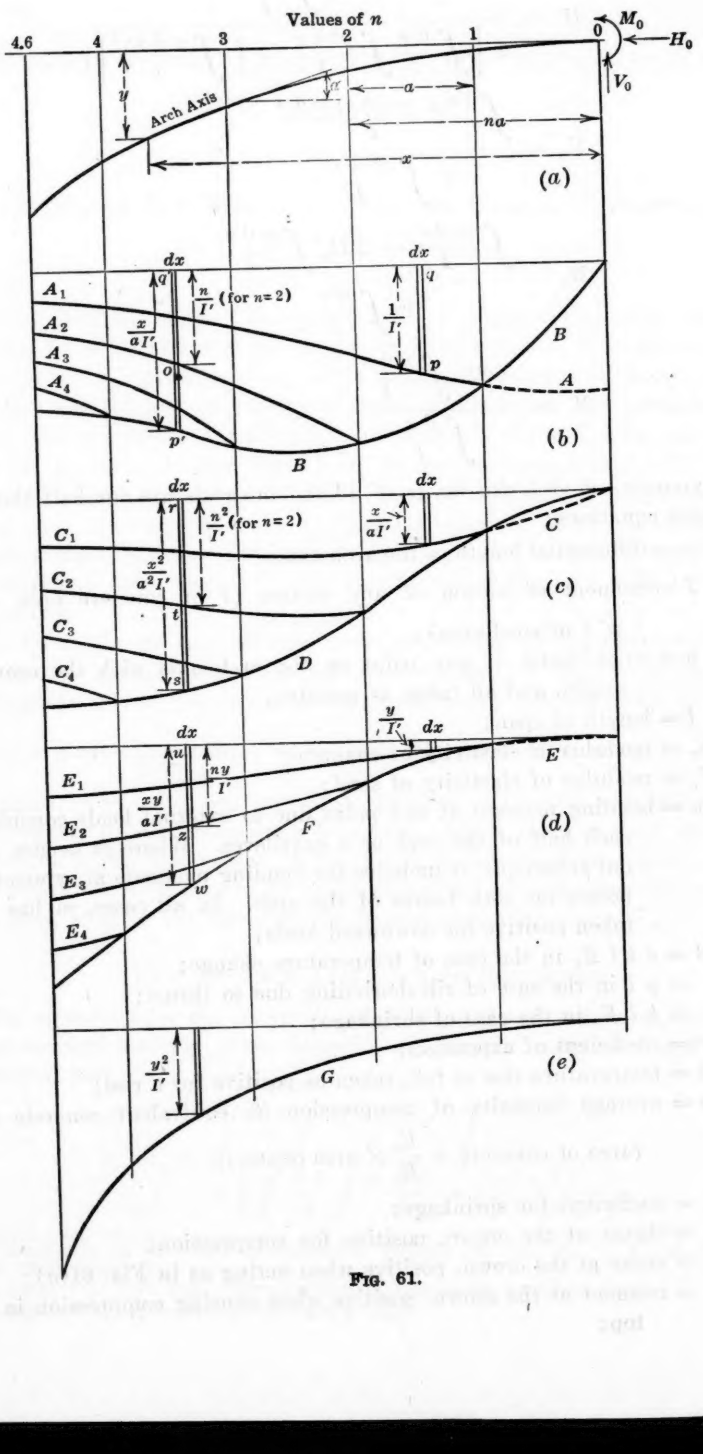
p = average intensity of compression on equivalent concrete area (area of concrete + $\frac{E_s}{E_c} \times$ area of steel);

k = coefficient for shrinkage;

H_0 = thrust at the crown, positive for compression;

V_0 = shear at the crown, positive when acting as in Fig. 61(*a*);

M_0 = moment at the crown, positive when causing compression in the top;



Values of n
4.6
4
3
2
1
0
M0
H0
V0
Arch Axis
y
a
na
x
(a)
dx
q
n/I' (for n=2)
A1
A2
A3
A4
x/aI'
o
p'
B
1/I'
p
A
(b)
r
n²/I' (for n=2)
C1
C2
C3
C4
x²/a²I'
t
s
D
x/aI'
C
(c)
u
ny/I'
E1
E2
E3
E4
xy/aI'
z
w
F
y/I'
E
(d)
y²/I'
G
(e)

FIG. 61.

H' = thrust at crown due to temperature change or rib-shortening;
M' = moment at crown due to rib-shortening or temperature change;
α = angle between tangent to arch axis at any point and the horizontal.

As $ds = \frac{dx}{\cos \alpha}$, the term, $\frac{ds}{I}$, in the equations may be replaced by $\frac{dx}{I'}$, in which $I' = I \cos \alpha$. Note that α is also the angle which the radius of curvature at any point makes with the vertical.

As influence values are sought, the value of m due to a unit load may be replaced by $(x - n\,a)$ for values of x greater than $n\,a$. For values of x less than $n\,a$, $m = 0$. Considering the left half of the arch,

$$(m_L - m_R) = (x - n\,a),$$

as $m_R = 0$.

Procedure.—The arch is first divided into any convenient number, n, of equal horizontal divisions each a in length. It is not necessary that a should be an aliquot part of the span, but it is somewhat simpler if a is a number of even feet in length. When a can be made 10 ft., some of the data can be reduced by shifting the decimal point.

Values of $\frac{1}{I'}$, $\frac{y}{I'}$, and $\frac{y^2}{I'}$ are computed for each of the n sections. These are all the preliminary computations needed. The steps in the analysis then are:

(1a).—Plot the values of $\frac{1}{I'}$ as ordinates with corresponding values of n as abscissas and connect these points with a smooth curve, A, Fig. 61 (b). The area of a strip, $p\,q$, is then $\frac{dx}{I'}$, and the area between Curve A and the axis of ordinates is equal to $\int \frac{dx}{I'}$.

(1b).—Plot n times the value of $\frac{1}{I'}$ on the same base at each corresponding value of n and connect the points thus found with a smooth curve, B, Fig. 61 (b). Since the ordinates of Curve B are values of $\frac{n}{I'}$, the area of a strip, $p'\,q'$, is $\frac{n\,dx}{I'}$, and as $n\,a = x$, this area is $\frac{x\,dx}{a\,I'}$. Thus, the area between Curve B and the axis is $\frac{1}{a} \int \frac{x\,dx}{I'}$.

(1c).—Connect all points representing $\frac{2}{I'}$, $\frac{3}{I'}$, $\frac{4}{I'}$, etc., beginning at Curve B and extending to the left to the springing line, giving Curves A_1 to A_4, Fig. 61 (b).

Assume a unit load placed at $n = 2$; the value of m at any point, x, is 1 $(x - n\,a)$ which may be written $a \left(\frac{x}{a} - n\right)$, and the value of $\frac{m\,dx}{I'}$ may be

expressed as $a\left(\frac{x}{a} - n\right)\frac{d\,x}{I'}$. Since the length of the strip, $o\,q'$, is $\frac{2}{I'}$ and of $p'\,q'$, $\frac{x}{a\,I'}$, the length of $p'\,o$ is $\left(\frac{x}{a} - n\right)\frac{1}{I'}$, and the area between Curve B and Curve A_2 is $\frac{1}{a}\int\frac{m\,d\,x}{I'}$ for a unit load at $n = 2$, the limits being from $n = 2$ to the springing line.

For a unit load at any other value of n, the value of $\frac{1}{a}\int\frac{m\,d\,x}{I'}$ may be found in the same way by evaluating the area between Curve B and the Curve A_n.

(2a).—With a new base, Fig. 61 (c), plot Curve B using a smaller scale for the ordinates, which will locate Curve C, giving values of $\frac{n}{I'}$.

(2b).—Plot n times these values of $\frac{n}{I'}$ at each corresponding value of n and connect the points thus found with a smooth curve, D, Fig. 61 (c).

Since the ordinates of Curve D are values of $\frac{n^2}{I'}$, the area of a strip, $r\,s$, is $n^2\frac{d\,x}{I'}$, and as $n\,a = x$, this area is equal to $x^2\frac{d\,x}{a^2\,I'}$. Thus, the area between Curve D and the axis is $\frac{1}{a^2}\int x^2\frac{d\,x}{I'}$.

(2c).—Connect all points representing $\frac{2\,n}{I'}$, $\frac{3\,n}{I'}$, $\frac{4\,n}{I'}$, etc., beginning at Curve D and extending to the left to the springing line, giving Curves C_1 to C_4, Fig. 61 (c).

Assume a unit load placed at $n = 2$. As in Item (1c), the value of m is $a\left(\frac{x}{a} - n\right)$. Hence,

$$m\,x = a\,x\left(\frac{x}{a} - n\right) = a^2\left(\frac{x^2}{a^2} - n^2\right)$$

and,

$$m\,x\,\frac{d\,x}{I'} = a^2\left(\frac{x^2}{a^2} - n^2\right)\frac{d\,x}{I'}$$

$x^2\frac{d\,x}{a^2\,I'}$ is the area of the strip, $r\,s$, and $n^2\frac{d\,x}{I'}$ is the area of the strip, $r\,t$, when $n = 2$. Hence, the area of the strip, $t\,s$, is:

$$\left(\frac{1}{a^2}\right)\left(m\,x\,\frac{d\,x}{I'}\right),$$

and the area between Curve D and Curve C is equal to:

$$\left(\frac{1}{a^2}\right)\int m\,x\,\frac{d\,x}{I'}$$

the limits being from $n = 2$ to the springing line.

For a unit load at any other value of n, the value of:

$$\left(\frac{1}{a^2}\right)\int m\,x\,\frac{d\,x}{I'}$$

may be found in the same way by evaluating the area between Curve D and Curve C_n.

(3*a*).—Plot the values of $\frac{y}{I'}$ as ordinates from a third base, using the corresponding values of n as abscissas, and connect these points with a smooth curve, E, in Fig. 61 (*d*). The area between this curve and the axis of ordinates is equal to $\int \frac{y\,d\,x}{I'}$.

(3*b*).—Plot n times the value of $\frac{y}{I'}$ on the same base at each corresponding value of n and connect the points thus found with a smooth curve, F, Fig. 61 (*d*).

(3*c*).—Connect all points representing $\frac{2\,y}{I'}$, $\frac{3\,y}{I'}$, $\frac{4\,y}{I'}$, etc., beginning at Curve F and extending to the left to the springing line, giving Curves E_1 to E_4, Fig. 61 (*d*).

Assume a unit load at $n = 2$. The value of $m\,y$ is then:

$$y\,(x - n\,a), \text{ or } a\,y\left(\frac{x}{a} - n\right)$$

and,

$$m\,y\,\frac{d\,x}{I'} = a\,y\left(\frac{x}{a} - n\right)\frac{d\,x}{I'}$$

The area of the strip, $u\,w$, is $\left(\frac{x\,y}{a\,I'}\right)d\,x$, and that of the strip, $u\,z$, is $n\,y\,\frac{d\,x}{I'}$. Hence, the area of the strip, $z\,w$, is $\frac{1}{a}\left(m\,y\,\frac{d\,x}{I'}\right)$ and the area between Curves F and $E_2 = \frac{1}{a}\int m\,y\,\frac{d\,x}{I'}$ for a unit load at $n = 2$.

For a unit load at any other value of n, the value of $\frac{1}{a}\int m\,y\,\frac{d\,x}{I'}$ may be found in the same way by evaluating the area between Curve F and Curve E_n.

(4).—Plot the values of $\frac{y^2}{I'}$ as ordinates on a fourth base, Fig. 61 (*e*). The area between this curve and its axis of ordinates is the $\int y^2\,\frac{d\,x}{I'}$.

(5).—The values obtained by evaluating the various areas in Items (1) to (4) may now be entered in a table similar to Table 26. The remainder of the work of obtaining the influence values for H_0, V_0, and M_0 is indicated at the top of the columns of this table, the figures in parentheses referring to the steps in the foregoing outline.

(6).—Computing Δ involves no difficulty, and as all the other terms in the equation for H' have already been found, the crown thrust and moment due to temperature or rib-shortening are readily obtained by substituting in the expressions for these quantities.

TABLE 26.—COMPUTATIONS IN SOLVING FOR H_0, V_0 AND M_0 FOR INFLUENCE VALUES.

n	(2c)	(2b) V_0	(3c)	(1c)	(1a)	(3a)	H_0				M_0
	$\frac{1}{a^2}\int\frac{m\,x\,d\,x}{I'}$	$\frac{\text{Column (1)}}{2 \times 116.7}$	$\frac{1}{a}\int\frac{m\,y\,d\,x}{I'}$	$\frac{1}{a}\int\frac{m\,d\,x}{I'}$	$13.3\,a \times$ Column (3)	$55.2\,a \times$ Column (4)	Column (5) − Column (6)	$\frac{\text{Column (7)}}{10\ 520}$	$a \times$ Column (4)	$55.2 \times 2\,H_0$	Column (9) − Column (10)	$\frac{\text{Column (11)}}{2 \times 13.3}$
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
0	116.7	0.500	252	31.4	33 500	17 300	16 200	1.54	314	170	144	−5.41
1	71.5	0.320	167	15.4	22 200	8 500	13 700	1.30	154	143	11	−0.41
2	36.1	0.161	960	6.1	12 750	8 360	9 390	0.89	61	98	−37	−1.39
3	12.0	0.054	400	1.6	5 320	880	4 440	0.42	16	47	−31	−1.17
4	1.8	0.008	80	0.2	1 060	110	950	0.09	2	10	− 8	−0.30
4.6	0.0	0.000	0	0.0	0	0	0	0.00	0	0	0	0.00

NOTES.—
$a = 10$

Column (2): $\frac{1}{a^2}\int\frac{x^2\,d\,x}{I'} = 116.7$; Column (5): $\int\frac{d\,x}{I'} = 13.3$

Column (6): $\int\frac{y\,d\,x}{I'} = 55.2$; From Fig. 61 (e): $\int\frac{y^2\,d\,x}{I'} = 624$

Column (8): $2\left[\int\frac{y^2\,d\,x}{I'}\int\frac{d\,x}{I'} - \left(\int\frac{y\,d\,x}{I'}\right)^2\right] = 2\left[624 \times 13.3 - (55.2)^2\right] = 10\ 520$

PART II.—ARCH ANALYSIS WITH INTEGRAPH

A modification of the method outlined in Part I which will simplify the work still further can be applied when an integraph is available. In the fundamental equations given at the beginning of Part I, the value of m for a unit load at a distance, $n\,a$, from the origin is $x - n\,a$, and the integrals in which m occurs are integrated between the limits, $n\,a$ and $\frac{l}{2}$ (since for values of x less than $n\,a$, m vanishes).

Then the various integrals involving m in the following forms may be written:

$$\int\frac{m\,d\,s}{I} = \int\frac{x\,d\,x}{I'} - n\,a\int\frac{d\,x}{I'} = a\left[\int\frac{x\,d\,x}{a\,I'} - n\int\frac{d\,x}{I'}\right] \quad \ldots(65)$$

$$\int\frac{m\,x\,d\,s}{I} = \int\frac{x^2\,d\,x}{I'} - n\,a\int\frac{x\,d\,x}{I'} = a^2\left[\int\frac{x^2\,d\,x}{a^2\,I'} - n\int\frac{x\,d\,x}{a\,I'}\right] \quad .(66)$$

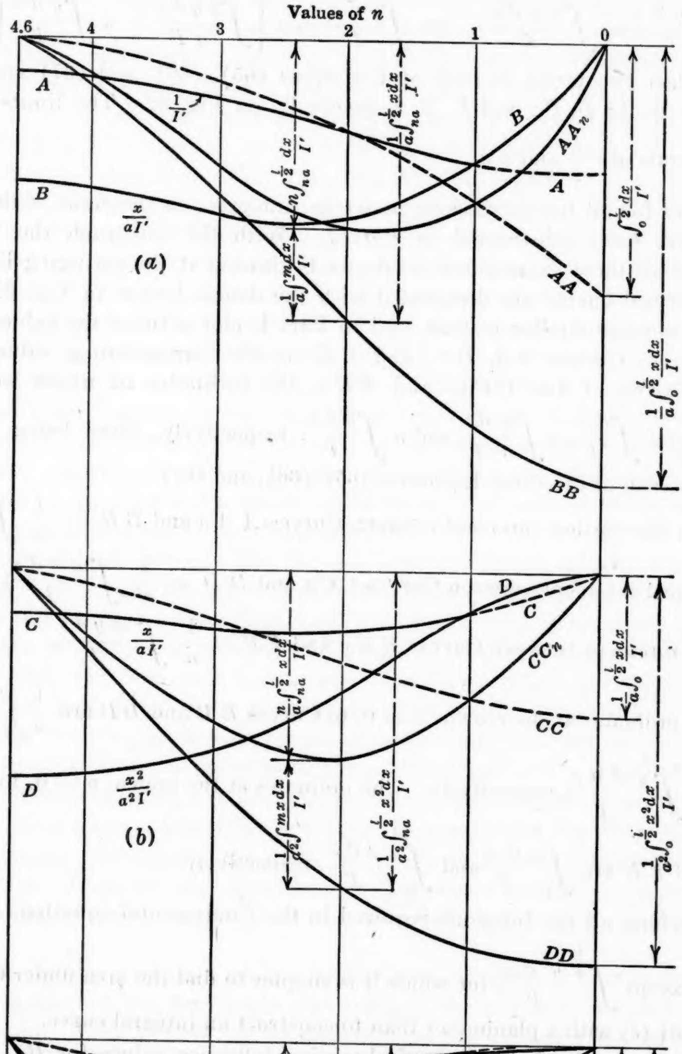


FIG. 62.

$$\int \frac{m\,y\,d\,s}{I} = \int \frac{x\,y\,d\,x}{I'} - n\,a \int \frac{y\,d\,x}{I'} = a \left[\int \frac{x\,y\,d\,x}{a\,I'} - n \int \frac{y\,d\,x}{I'} \right] \quad ..(67)$$

The last two terms in each of Equations (65), (66), and (67) are designated as B, A; D, C; and F, E, respectively, on Fig. 62. The limits for all the integrals are $\frac{l}{2}$ and $n\,a$.

Curves for all the expressions under the integrals on the right of the equations have been constructed in Part I. With the integraph the integral curves of all these expressions are drawn beginning at the springing line, and these integral curves are designated with the double letters in Fig. 62.

In a manner similar to that used in Part I, plot n times the values of the ordinates to Curves $A\,A$, $C\,C$, and $E\,E$, at the corresponding values of n, giving Curves $A\,A\,n$, $C\,C\,n$, and $E\,E\,n$, the ordinates of which would be equal to: $n \int \frac{d\,x}{I'}$, $n \int \frac{x\;dx}{a\;I'}$, and $n \int \frac{y\,d\,x}{I'}$, respectively, these being the last terms in each of the three Equations (65), (66), and (67).

Then the vertical intercept between Curves $A\,A\,n$ and $B\,B = \frac{1}{a} \int \frac{m\;d\,s}{I}$; the vertical intercept between Curves $C\,C n$ and $D\,D = \frac{1}{a^2} \int \frac{m\;x\;d\;s}{I}$; and the vertical intercept between Curves $E\,E\,n$ and $F F = \frac{1}{a} \int \frac{m\,y\,d\,s}{I}$.

The ordinates at the crown, $n = 0$, to Curves $B\,B$ and $D\,D$ are $\frac{1}{a} \int_0^{\frac{l}{2}} \frac{x\;d\;x}{I'}$ and $\frac{1}{a^2} \int_0^{\frac{l}{2}} \frac{x^2\,d\,x}{I'}$, respectively. The ordinates at the crown, $n = 0$, to Curves $A\,A$ and $E\,E$ are $\int_0^{\frac{l}{2}} \frac{d\,x}{I'}$ and $\int_0^{\frac{l}{2}} \frac{y\,d\,x}{I'}$, respectively.

Therefore, all the integrals required in the fundamental equations are now found except $\int_0^{\frac{l}{2}} \frac{y^2\,d\,x}{I'}$, for which it is simpler to find the area under Curve G of Fig. 61 (e) with a planimeter than to construct an integral curve.

The remainder of the work of obtaining influence values for H_0, V_0, and M_0 can best be done in tabular form as in Part I.

E. H. Harder,* Assoc. M. Am. Soc. C. E. (by letter).†—The writer has used the method and tables of Strassner for several years with considerable success and is glad to see this praiseworthy work translated into English. If a hingeless symmetrical concrete arch is designed according to the shape and form computed by this method, its tables or curves are accurate and can be used with confidence; but the writer disagrees with the author's statements that they can be used without a thorough knowledge of the entire theory of

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† Received by the Secretary, February 6, 1925.

indeterminate arches. The writer is quite certain that Strassner never intended his tables to be used by a novice. They are invaluable, however, to a mature designer who has designed such arches according to more laborious methods and who may then expect to have his work made a little pleasanter. For ordinary earth-filled arches, these tables may be readily used, but for large spans with transverse spandrel walls or columns, the novice could not be expected to know their limitations or how to make the proper allowances.

Beginning with Section IV, "The Arch Axis",* the author gives the mathematical derivation of the equation for a curve known as a transformed catenary. This curve is a linear arch for a load of uniform density between a horizontal roadway and the curve itself. The linear arch for a uniform horizontal load is, of course, a parabola. For filled arches with a horizontal roadway, the transformed catenary is the correct curve to use, but for large open spandrel arches a departure from this curve is often necessary.

Judging from a number of textbooks which have been written, one gains the impression that any reasonable curve (segment of a circle, parabola, or ellipse) is the correct curve for the arch axis provided one later blindly applies the elastic theory to determine the moments and thrusts. For live loads, this procedure is not questioned, but for dead loads, it is inaccurate and decidedly questionable in the minds of some engineers.

The dead load pressure line for any well designed concrete arch should coincide exactly with the center line of the arch. For earth-filled arches with a horizontal roadway such a dead load pressure line is a transformed catenary. For other conditions of loading the shape of the arch center line may be slightly different from such a catenary. If this condition has been complied with, then, neglecting for a moment arch-shortening, there will be uniform stress over every radial section passed through the arch. Due, however, to the elastic properties of the material of the arch, moments of a secondary nature will be caused by what is known as arch-shortening, which is clearly treated in the author's text. Any other shape for the arch center line will cause bending moments throughout the arch in addition to those caused by arch-shortening. There are reasons to believe that such moments are questionable; whether questionable or not, they may seriously impair the economy of the arch.

Some writers (Tolkmit, Melan, and others) advocate a transformed catenary designed for the dead loads plus one-half the uniform live load over the entire span. Such a curve can be computed by the method given in the paper by adding $\frac{1}{2}\,\frac{1}{p}$ to w_s and w_c, respectively. The writer, however, prefers the first method.

To eliminate the secondary moments due to arch-shortening has been the aim of engineers since the beginning of the construction of modern concrete arches. In recent years several writers have sought to accomplish this by a further modification of the transformed catenary or "line-of-thrust" arch, that is, by a change of curvature. In general, such a change means a

* *Proceedings*, Am. Soc. C. E., November, 1924, Papers and Discussions, p. 1362.

flatter curve at the crown and a sharper curve at the springing line. Two interesting papers on this phase of the subject have been written by Gerhard Neumann* and Professor A. Ostenfeld,† of Copenhagen, Denmark. The latter paper is a mathematical jugglery which gives excellent results on paper, but the writer has been unable to conceive how a secondary moment can be overcome by a mere change of shape. Arch-shortening thrusts are produced by an elastic deflection and such thrusts can only be eliminated or neutralized by external forces or moments producing similar deflections in the reverse direction. In the first edition of his book, Strassner‡ also gives a method for improving the shape of the arch axis but, in a later edition, this "improvement" is omitted.

The earliest recognized methods of eliminating the dead load arch-shortening moments called for temporary hinges at the crown and springing lines of the arch. After this three-hinged arch carried all the dead loads, the hinges were closed and thereafter the arch acted as a hingeless structure for live loads, temperature changes, shrinkage, and additional abutment displacements. This method is practical but expensive, especially for long heavy ribs. Few typical examples of such construction methods are recorded. In 1908, the Stubenrauchbrücke was built in Berlin, Germany. The central span was a 200-ft. steel arch flanked on each end by 64-ft. concrete arches. These concrete arches had temporary hinges at the crowns and springing lines and after the entire dead load of all three spans had been placed, the hinges of the concrete arch were closed. In 1921, a ribbed arch of 135 ft. span with temporary hinges was built in Warrington, England.§

A newer method developed by Buchheim and Heister in Germany and by Freysinnet in France makes use of hydraulic jacks inserted between two planes which cut the crown of the arch. In this manner a horizontal thrust is created which forces the two crown sections apart and lifts the arch from its centering. At the same time, careful measurements are made from which it is possible to determine accurately the modulus of elasticity of the entire structure. This modulus is necessary in order to make the proper allowances for arch-shortening, temperature, shrinkage, and abutment displacements. After the centering has been removed, the pressure on the hydraulic jacks is reduced to the predetermined amount and the crown section is closed. Although this method ("Gewölbe Expansions Verfahren") was first developed to repair arch bridges which had suffered damage from abutment displacements,‖ it has been used in the erection of arches in Germany¶ and France.** In the latter country, this method has been developed to such an extent by Freysinnet as to make the economical construction of long-span bridges possible.

* *Beton und Eisen*, No. XIII (1922).
† *Beton und Eisen*, No. XIII (1923).
‡ "Neuere Methoden zur Statik der Rahmentragwerke und der elastischen Bogentraeger."
§ *Le Genie Civil*, July 30, 1921.
‖ "Handbuch für Eisenbeton", Vol. VII.
¶ "Handbuch für Eisenbeton", Vol. I.
** *Engineering News-Record*, September 18, 1924.

Besides eliminating the distressing dead load arch-shortening moments, it also eliminates the uncertainties of the location of the dead load pressure line. Closure of the crown is made at a definite temperature and thereby another indeterminate quantity in the present methods of constructing reinforced concrete arches is removed. Without the use of this method, it is impossible to know the temperature which creates no moments in the structure.

In Section V, "Form of Arch Rib",* is given the equation of a special case of the law of variation of the moments of inertia throughout the arch rib. The general equation is,

$$\frac{I_c}{I_x \cos \phi} = 1 - (1 - m)\, Z^v$$

in which, the exponent, v, may vary from 0 to 2. In his work, Strassner has chosen v equal to 1 purely for the sake of simplifying those integrations which involve the variable arch moments of inertia.

Little has been written about the subject of the variation of the moments of inertia along the arch axis and wherever it has appeared, it has been treated very superficially. Generally, arbitrary assumptions were made (as, for example, that $I_x \cos \phi$ was constant) or an arbitrary set of proportions were given for the arch depths which proportions were dependent on the depths at the crown and at the haunch. The general law just given was the result of a research which had for its purpose the distribution of the arch moments of inertia so that the maximum stresses would be equal for all sections along the arch axis. If such an arch could be designed, it would be more economical, of course, than an arch of haphazard form wherein the crown stresses might be twice the haunch stresses, or *vice versa*.

In Fig. 63 (*a*) are plotted the maximum positive and negative live load moments of a large hingeless reinforced concrete arch designed according to this method. This curve is typical of all hingeless arches and shows that besides the crown section, there is another section of maximum moments at seven-tenths of the length of the half span measured from the crown. In Fig. 63 (*b*) is a graph of the dead load arch-shortening moments, in Fig. 63 (*c*), a graph of moments caused by temperature changes, and in Fig. 63 (*d*), a graph showing the combination of all moments of Fig. 63 (*a*), (*b*), and (*c*). In Fig. 63 (*d*) are also shown two moment curves which are proportional to the moments of inertia of sections along the arch axis. Curve *a* was computed according to the method given in the paper and Curve *b* according to the general law with $v = 2$.

Fig. 63 (*e*) is a graph of the maximum fiber stresses for dead and live load, arch-shortening, and temperature changes. The fiber stresses at the haunch are quite close to the maximum crown stresses, but in the region of the minimum moments shown in Fig. 63 (*d*), they are below the allowable maximum. If an arch were designed so that the stresses would neither be greater nor less than the allowable maximum stress, there would result a curious structure which would increase in thickness from the crown to about four-tenths the half span, then decrease to a minimum thickness at

* *Proceedings*, Am. Soc. C. E., November, 1924, Papers and Discussions, p. 1373.

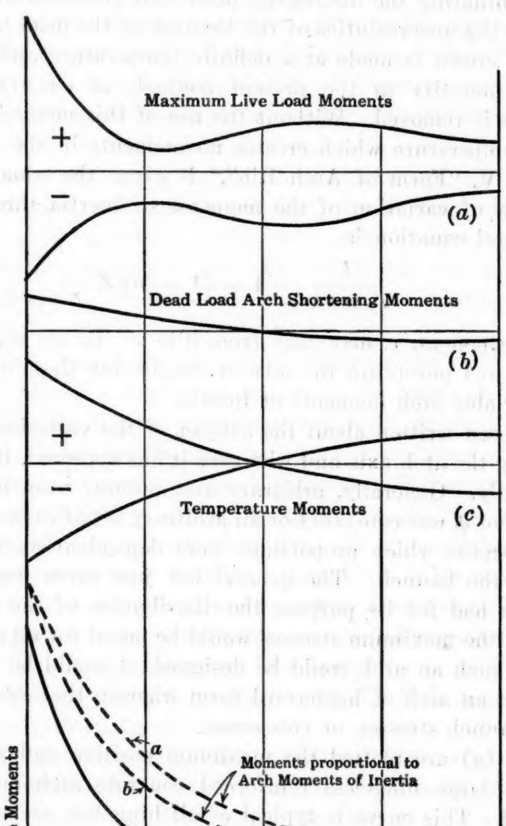
Maximum Live Load Moments
+
−
(a)
Dead Load Arch Shortening Moments
(b)
+
−
Temperature Moments
(c)
a
b
Moments proportional to
Arch Moments of Inertia
Positive Moments
Negative Moments
+
−
Combined Moments
(d)
Stress
Lower Fibre
Upper Fibre
(e)
l/2
Haunch
Quarter Point
Crown

FIG. 63.

about seven-tenths the half span and from that point the arch haunch would have a rapid increase in thickness to the maximum section needed at that point. Such an arch would have an ungainly appearance, and would be unacceptable from an æsthetic point of view no matter what its advantages might be.

The author's method for determining the thickness of arch rib* is very unsatisfactory and is undoubtedly based on the first edition of Strassner's book. In a later edition, he has greatly improved the chapter on the planning or designing of the arch rib.

Equation (46)† is that of the dead load horizontal thrust and may be written:

$$H_d = C_d \frac{w_c\, l^2}{r}$$

in which,

$$C_d = \frac{g - 1}{4\, K^2}$$

(see Table 9‡):

The empiric equation for C_d is given by Strassner:

$$C_d = 0.1080 + 0.0190\, g - 0.0005\, g^2$$

For uniform live loads:

$$H_l = C_l \frac{p\, l^2}{r},$$

in which, for $m = 0.3$:

$$C_l = 0.0579 + 0.0035\, g$$

and

$$M_l = K_l\, pl^2$$

in which, for $m = 0.3$:

$$K_l = 0.00426 + 0.00037\, g$$

These coefficients may be taken from Figs. 50 to 55,§ but may also be expressed by empiric equations. Strassner gives a list of such equations for various values of m, but as the value of $m = 0.3$ is most prevalent, the writer will use only those equations for this value to derive general formulas for crown and haunch thicknesses:

$$\cos \phi_s = \frac{1}{\sqrt{1 + \tan^2 \phi_s}} = \frac{1}{\sqrt{1 + x \left(\frac{r}{l}\right)^2}}$$

in which,

$$x = 11.50 + 4.55\, g - 0.13\, g^2$$

For a change of temperature:

$$H_t = \frac{t\, \alpha\, l}{\int y^2\, d\, w} = \frac{t\, \alpha\, E\, I_c}{C\, r^2}$$

* *Proceedings*, Am. Soc. C. E., November, 1924, Papers and Discussions, p. 1393.

† *Loc cit.*, p. 1368.

‡ *Loc. cit.*, p. 1372.

§ *Loc. cit.*, pp. 1422–1424.

in which, $I_c = \frac{b\, d_c^{\,3}}{12}$ for a rectangular cross-section. Then, for $b = 1.0$:

$$H_t = \frac{E\, \alpha\, t\, d^3}{12\, C\, r^2} = C_t \frac{t\, d_c^{\,3}}{r^2}$$

$$M_t = -H_t \,.\, y = \frac{E\, \alpha\, t\, d_c^{\,3}}{12\, C\, r} \times \frac{y}{r} = K_t \frac{t\, d_c^{\,3}}{r}$$

C may be obtained from Table 18 * and $\frac{y}{r}$ from Table 6. †

For $m = 0.3$, these coefficients may be expressed empirically:

$$C_t = 4218 + 136.8\, g$$
$$K_t = -1037.1 + 4.56\, g$$

Dead load rib-shortening is expressed by:

$$H_{RS} = -\, u'\, H_d$$

in which,

$$u' = \frac{l}{E\, A_m'\, \int y^2\, d\, w}$$

Proceeding as before:

$$H_{RS} = C_e \frac{w_c\, l^2\, d_c^{\,2}}{r^3}$$

and the section moment is:

$$M_{RS} = K_e \frac{w_c\, l^2\, d_c^{\,2}}{r^2}$$

in which,

$$C_e = \frac{C_d}{12\, C\, C_m'} \text{ and } K_e = C_e \frac{y}{r}$$

For $m = 0.3$:

$$C_e = -\,0.155 - 0.033\, g$$
$$K_e = 0.0408 + 0.0046\, g$$

The fiber stresses may be computed by the usual formula for homogeneous sections without reinforcement:

$$f_c = \frac{T}{b\, d} \pm \frac{6\, M}{b\, d^2}$$

Then, for a rib, 1 ft. wide:

$$f_c = \frac{T}{d} \pm \frac{6\, M}{d^2}$$

Substituting the terms derived previously:

$$T \,.\, \cos\phi = C_d \frac{w_c\, l^2}{r} + C_l \frac{p\, l^2}{r} + C_t \frac{t\, d_c^{\,3}}{r^2} + C_e \frac{w_c\, l^2\, d_c^{\,2}}{r^3}$$

and

$$M = K_l \cdot p\, l^2 + K_t \frac{t\, d_c^{\,3}}{r} + K_e \frac{w_c\, l^2\, d_r^{\,2}}{r^2}$$

* *Proceedings*, Am. Soc. C. E., November, 1924, Papers and Discussions, p. 1387.
† *Loc cit.*, p. 1369.

Substituting these equations into the equation for f_c for the crown section:

$$f_c = C_1 \frac{l^2}{d_c\, r} + C_2 \frac{d_c^{\,2}}{r^2} + C_3 \frac{l^2\, d_c}{r^3} \pm 6 \left(K_1 \frac{l^2}{d_c^{\,2}} + K_2 \frac{d_c}{r} + K_3 \frac{l^2}{r^2}\right)$$

in which,

$$C_1 = C_d \cdot w_c + C_l \cdot p\,;\quad C_2 = C_t \cdot t\,;\quad C_3 = C_e \cdot w_c$$
$$K_1 = K_l \cdot p\,;\qquad K_2 = K_t \cdot t\,;\quad K_3 = K_e \cdot w_c$$

This equation can be further simplified by omitting the two terms depending on C_2 and C_3. These two terms are expressions for the temperature change and dead load arch-shortening. For maximum stresses at the crown one must consider a drop in temperature which has the same sign and effect as the dead load arch-shortening. For simplification, assume both to be equal percentages of the factors, K_2 and K_3. All equations for the crown stresses, therefore, become:

$$\left.\begin{aligned} f_{c\,(\text{upper})} &= C_1 \frac{l^2}{d_c\, r} + 6\, K_1 \frac{l^2}{d_c^{\,2}} + 6\,(1 - \psi_c) \left(K_2 \frac{d_c}{r} + K_3 \frac{l^2}{r^2}\right) \\ f_{c\,(\text{lower})} &= C_1 \frac{l^2}{d_c\, r} - 6\, K_1 \frac{l^2}{d_c^{\,2}} - 6\,(1 + \psi_c) \left(K_2 \frac{d_c}{r} + K_3 \frac{l^2}{r^2}\right) \end{aligned}\right\} \ldots (68)$$

Thus, although both these terms were omitted, they have been re-introduced through the factor, ψ_c, which varies between the values, 0.05 and 0.20, depending on the variation of m and the ratio of crown thickness to the rise.

For $m = 0.3$:

$$\psi_c = 0.80 \frac{d_c}{r}$$

The ruling fiber stress used in determining the crown thickness is that of the upper fiber. The equation for this stress is rewritten in the following form:

$$f_c = f_c' + 6\,(1 - \psi_c) \left(K_2 \frac{d_c}{r} + K_3 \frac{l^2}{r^2}\right) \ldots\ldots\ldots\ldots (69)$$

in which, f_c' is the uniform compression caused by the dead load and live load thrusts. Substituting Equation (69) for f_c (upper) in Equation (68) and solving for d_c:

$$d_c = \frac{C_1\, l^2}{2\, r\, f_c'} \left[1 + \sqrt{1 + \frac{24\, f_c'\, K_1}{\left(\frac{C_1\, l}{r^2}\right)^2}}\,\right] \ldots\ldots\ldots\ldots (70)$$

This equation is useful only to check a dimension for the crown previously estimated or guessed. This must first be done in order to obtain the factor, g, on which depends the shape of the arch center line. After obtaining all C and K coefficients, the value of f_c' is obtained from Equation (69) by substituting for f_c the allowable fiber stress for which the arch is to be designed. If this guess was correct, Equation (70) will prove it. If not, there will result a new and better value for d_c, which may be checked by Equation (70).

If the crown thickness is unknown and the designer has no means of finding how close his guess may be, there is still another and better method of determining the crown thickness.

$w_c = w_0 + \gamma\, d_e$, in which w_0 is the weight per foot of arch rib superimposed above the crown and y is the density of the arch material (150 lb. per cu. ft. for concrete):

$$\begin{aligned} C_1 &= C_d\,(w_0 + \gamma\, d_c) + C_l \cdot p \\ &= C_1' + C_1'' \cdot d_c \end{aligned}$$

Substitute this equation for C_1 in Equation (68). Then, the first term becomes:

$$C_1' \frac{l^2}{d_c\, r} + C_1'' \frac{l^2}{r}$$

in which,

$$C_1' = C_d \cdot w_0 + C_l \cdot p \quad \text{and} \quad C_1'' = C_d \cdot \gamma$$

K_3 will undergo a similar change and, as before:

$$f_{c\,(\text{upper})} = C_1' \frac{l^2}{d_c\, r} + C_1'' \frac{l^2}{r} + 6\, K_1 \frac{l^2}{d_c^{\,2}}$$

$$+ 6\,(1 - \psi_c)\left(K_2 \frac{d_c}{r} + K_3' \frac{l^2}{r^2} + K_3'' \frac{l^2}{r^2}\, d_c\right)$$

$$f_{c\,(\text{lower})} = C_1' \frac{l^2}{d_c\, r} + C_1'' \frac{l^2}{r} - 6\, K_1 \frac{l^2}{d_c^{\,2}}$$

$$- 6\,(1 + \psi_c)\left(K_2 \frac{d_c}{r} + K_3' \frac{l^2}{r^2} + K_3'' \frac{l^2}{r^2}\, d_c\right)$$

in which the various coefficients are those previously given and $K_3' = K_e \cdot w_0$ and $K_3'' = K_e \cdot \gamma$.

An example, using the author's arch, will illustrate the use of these two formulas:

$$l = 100 \text{ ft.} \qquad m = 0.339$$
$$r = 15 \text{ ft.} \qquad g = 4.70.$$

$$C_d = 0.1080 + 0.0190 \times 4.70 - 0.0005 \times 4.70^2 = 0.1863$$
$$K_l = 0.00426 + 0.00037 \times 4.70 = 0.0060$$
$$C_l = 0.0579 + 0.0035 \times 4.70 = 0.0744$$
$$K_t = -1\,037.1 + 4.56 \times 4.70 = -1\,015.7$$
$$K_e = 0.0408 + 0.0046 \times 4.70 = 0.0624$$
$$\psi_c = \frac{0.80 \times 1.5}{15} = 0.08$$

$$C_1' = 0.1863 \times 280^* + 0.0744 \times 200 = 67.04$$
$$C_1'' = 0.1863 \times 150 = 27.95$$
$$K_1 = 0.0060 \times 200 = 1.20$$
$$K_2 = -1\,015.7 \times (-35) = 35\,549.50$$
$$K_3' = 0.0624 \times 280 = 17.47$$
$$K_3'' = 0.0624 \times 150 = 9.36$$

* *Proceedings*, Am. Soc. C. E., November, 1924, Papers and Discussions, p. 1371.

$$f_{c\,(\text{upper})} = \frac{67.04 \times 100^2}{d_c \times 15} + \frac{27.95 \times 100^2}{15} + \frac{6 \times 1.20 \times 100^2}{d_c^{\,2}}$$

$$+ 5.52 \left(\frac{35\,549.5 + d_c}{15} + \frac{17.47 \times 100^2}{15^2} + \frac{9.36 \times 100^2 \times d_c}{15^2} \right)$$

$$f_{c\,(\text{lower})} = \frac{67.04 \times 100^2}{d_c \times 15} + \frac{27.95 \times 100^2}{15} - \frac{6 \times 1.20 \times 100^2}{d_c^{\,2}}$$

$$- 6.48 \left(\frac{35\,549.5\, d_c}{15} + \frac{17.47 \times 100^2}{15^2} + \frac{9.36 \times 100^2 \times d_c}{15^2} \right)$$

$$f_{c\,(\text{upper})} = 22\,900 + 15\,890\, d_c + \frac{44\,700}{d_c} + \frac{72\,000}{d_c^{\,2}}$$

$$f_{c\,(\text{lower})} = 13\,580 - 18\,700\, d_c + \frac{44\,700}{d_c} - \frac{72\,000}{d_c^{\,2}}$$

All these quantities are in foot-units, therefore the stresses will be pounds per square foot, which may be divided by 144 to obtain pounds per square inch. By substituting various values of d_c in these equations, the values given in Table 27 will be obtained.

TABLE 27.—STRESSES IN ARCH RING FOR VARIOUS CROWN DEPTHS.

Crown depth, in feet.	$f_{c\,(upper)}$, in pounds per square inch.	$f_{c\,(lower)}$, in pounds per square inch.
0.5	2 830	—1 350
1.0	1 080	— 225
1.5	753	— 116
2.0	660	— 135
2.5	639	— 186
3.0	649	— 247

These results are plotted in Fig. 64 and indicate that for a crown thickness of 18 in. with a uniform live load of 200 lb. per sq. ft. and a fall of temperature of 35° Fahr., an upper fiber stress of 753 lb. per sq. in. and a tension in the lower fiber of 116 lb. per sq. in. may be expected. For these conditions of loading and temperature, one would expect to obtain a minimum compression in the upper fiber for a crown thickness of 30 in., while a minimum tension in the lower fiber would be obtained for a crown thickness of 20 in.

Similar curves could be drawn for the springing line section, but such additional work would entail too much unnecessary work. For average conditions, $m = 0.3$, by which the springing thickness can be computed after choosing the crown thickness. If, then, it is desired to adhere to the same maximum compressive stress chosen for the crown, the following formula may be used to check the thickness d_s:

$$d_s = \frac{C_1\, l^2}{2 \cdot r \cdot f_c \cdot \cos \phi_s} \left[1 + \sqrt{1 - \frac{4 \cdot f_c \cdot A}{\left[\dfrac{C_1\, l}{r \cdot \cos \phi_s} \right]^2}} \right]$$

in which,

$$A = 6\, K_1 + 6 \left(1 - \frac{\psi_s}{\cos \phi_3} \right) \left(K_2 \frac{d_c^{\,3}}{l^2\, r} + K_3 \frac{d_c^{\,2}}{r^2} \right)$$

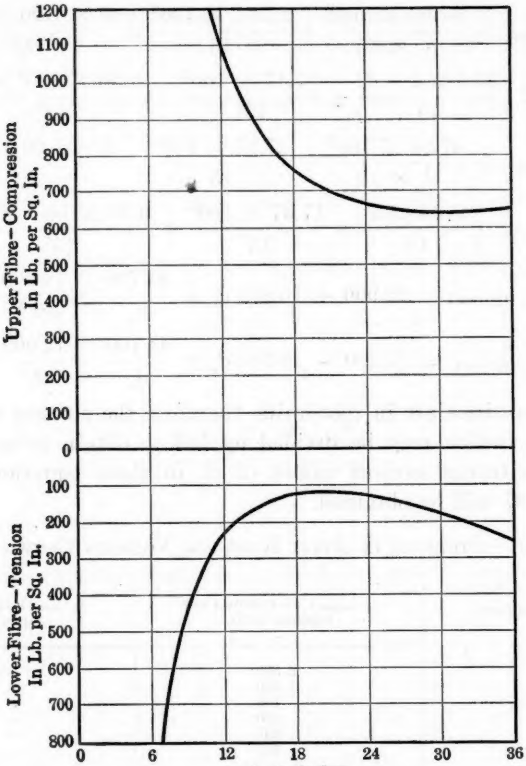

FIG. 64.

In these equations, the C and K coefficients must not be confused with those previously used for the crown thickness. For $m = 0.3$, these coefficients are:

$$C_d = \text{same as for crown}$$
$$K_l = -0.02100 + (0.00069 \times g)$$
$$C_l = 0.0356 - (0.0008 \times g)$$
$$K_t = 3\,159.2 + (146.9 \times g)$$
$$K_e = -0.1129 - (0.0280 \times g)$$
$$\psi_s = 0.22 \frac{d_s}{r}$$
$$C_1 = C_d \cdot w_c + C_l \cdot p$$
$$K_1 = K_l \cdot p$$
$$K_2 = K_t \cdot t^\circ$$
$$K_3 = K_e \cdot w_c$$

If the rib is widened toward the springing line, the constants, C_1, K_1, K_2, and K_3, must be reduced by the ratio of the crown width to the width at the springing line.

The formulas here presented have been used by the writer for a number of years and have assisted him greatly in the design of reinforced concrete arch bridges (at least, as far as the paper design is concerned); but the serious

student of such structures must often pause to wonder how close to reality his mathematical and computational efforts have come.

If the arch rib has been designed and executed as a line-of-thrust arch for the dead load, there remain the questions of arch-shortening and abutment displacements, shrinkage, and a possible internal re-organization of the arch due to "flow" or the "time-yield" of concrete, as it is most frequently called. There is a strong possibility that this latter property of concrete permits a readjustment of internal stresses and may even reduce temperature stresses, due to annual waves of temperature changes. At any rate, some mysterious forces are active in every arch, which cause readjustments of stresses, otherwise many arches now in use would collapse or, at best, be filled with cracks. The writer has had occasion to observe the crown deflections of several arches over a period of years and has come to the conclusion that if the stresses which mathematically correspond to these deflections were not relieved in some way, serious damage would be caused.

Then there is the question of temperature stresses. As previously mentioned, it is impossible to state definitely at what temperature there is no temperature thrust nor ensuing temperature moments. When the arch is concreted, sufficient heat is generated by the setting of the cement to raise the temperature of the mass of concrete. Temperatures as high as 128° Fahr. have been found in arches. If a large arch is poured in a few days in summer, what effect has such a rise of temperature when winter comes?

The writer also questions the uncertainty of the usual live load specifications for highway bridges. In the example, a live load of 200 lb. per sq. ft. was used for demonstration only. This load seems absurdly large for a 100 ft. span. Most proposed loadings would call for a uniform live load of about 100 lb. per sq. ft. for a span of that size. With four positions of the live load for maximum and minimum fiber stresses at any section, it stands to reason that a different loading is necessary for the hingeless arch than would apply to steel trusses.

With all these briefly mentioned uncertainties, the writer believes the present high designing stresses in use to be deplorable. In the past ten years or more, the Joint Committee on Standard Specifications for Concrete and Reinforced Concrete has reduced reinforced concrete column stresses, yet reinforced concrete arch rib stresses have been increased by individual engineers. This is due partly to the competition with steel bridges and partly to the engineer's desire to gain a reputation for designing cheap or economical bridges. The use of high unit stresses has resulted in the construction of stilt-like ribbed structures widely separated, with the roadway carried from rib to rib by long girders. At the crowns of the ribs, these long girders are usually rigidly connected to the arches, without any thought being given to the torsion developed in the ribs due to deflections in the floor-beams. Such structures may also be subjected to very high torsions due to wind pressures, yet no thought is expended on even investigating such stresses. Several years ago, in France, a reinforced concrete arch, celebrated for its slenderness, was destroyed during a wind storm.

In conclusion, the writer does not wish to appear as an alarmist although there are many reasons for feeling uneasy. He merely desires to point out that there were other things to consider aside from the mathematics of the elastic theory, things which the novice has not had the time nor the experience to consider.

CHARLES W. COMSTOCK,* M. AM. SOC. C. E. (by letter).†—The writer desires to record an emphatic protest against an expression occurring in the first sentence of this paper.

The term "elastic theory", which has unfortunately fallen into common use, is a careless and quite unnecessary short-cut for "mathematical theory of elasticity". It conveys a meaning very different than that intended, although often nearer the truth than the correct expression, the theory of elasticity having been frequently twisted and otherwise distorted beyond all recognition to meet some imaginary need for simplicity.

That the mathematical relationships on which engineers depend for calculated results are often complicated is no argument against their exactness or their use. Simplified formulas are much to be desired, but they can be justified only by detailed examination of each neglected or altered term and demonstration that its maximum value in any case is less than the allowable limit of error.

The theory of elasticity is by no means elastic. It is one of the most complete, consistent, and satisfactory in the whole range of mathematical physics within the limits set by the premises on which it is founded. These are simple, predicating only the proportionality of stress to strain and the property named elasticity, *viz.*, that the strained body will return to its original state upon removal of the stress. In many cases even the latter assumption is not a necessary prerequisite.

True, many problems lead to mathematical difficulties hitherto insuperable, and any presentation should be welcome that afforded solution of a problem which so far has proved baffling.

In examining the paper, the writer fails to find a new principle, a new result, the solution of a new problem, or even an improved solution of an old problem. By far the larger part of the text is devoted to the derivation of familiar formulas and the solution of problems. The remainder sets out a series of tables and diagrams the apparent intention of which is to standardize all arch proportioning in accordance with the author's views, leaving nothing to future designers but to take out the needed results as one would use a table of logarithms. The futility of such an effort should be evident. No two engineering problems are alike. Each presents its own peculiar features which must be studied as they appear and met with whatever resources and ingenuity the engineer can command. Cut and dried standards, even if they were feasible, dampen the engineer's enthusiasm, destroy his incentive, and kill any possibility of improvement in design or advancement in the art of construction.

The writer does not believe that the last word has been said on the calcula-

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† Received by the Secretary, February 10, 1925.

tion of stresses or the proportioning of structures. Until that time shall come, if ever, the greatest advancement will be made by encouraging each engineer to solve his problems in his own way with full knowledge of what others have done before. The duty of the Society and of its members is to keep the world informed of new problems, new solutions, and new discoveries.

In 1879 the scholarly Italian engineer Castigliano published his "Théorie de l'Equilibre des Systèmes Elastiques et ses Applications". No more concise, systematic, or exhaustive treatment of engineering uses of the theory of elasticity can be imagined. In the course of many years' study of such problems the writer has never encountered one that, if solvable at all, was not amenable to the principles and methods set forth in that masterly treatise, and without more labor than required by the so-called modern methods.

In the nearly half century which has elapsed since that classic publication there has been a veritable flood of textbooks, treatises, and so-called scientific papers which have purported to present something new, but which for the most part have recorded merely the mental contortions of the respective authors in grasping principles already fully developed.

The writer thinks the title of this paper is too comprehensive. "Design" consists in the choice of suitable materials and their proper arrangement to accomplish a given purpose. Stress calculation is a necessary step in this procedure, usually the simplest and often relatively unimportant. The other and more difficult problems rarely admit of unique solutions or mathematical determination. Much is left to the discretion and judgment of the engineer based on the experience of himself and others. If it were otherwise, engineers could be rated on the single standard of mathematical accomplishment, and any mathematician would be an engineer.

A. G. HAYDEN,* M. AM. SOC. C. E. (by letter).†—This discussion is pertinent to the paper only in a general way; and yet the writer feels that attention should be called to a particular line of experimental investigation that is much needed before full advantage can be taken of the great advances recently made in this field of engineering mathematics. What follows applies particularly to short-span arch bridges on the design of which heavy concentrated live loads have considerable influence.

It is generally recognized that certain practical problems await solution before accuracy of mathematical analysis becomes the governing factor in the design of such bridges. All methods of analysis presuppose ideal conditions of loading that are seldom realized in practice. Accuracy is possible for conditions of loading that approach uniformity of distribution, but in the case of heavy concentrations the designer is faced with the necessity of making certain out-and-out assumptions as to their lateral distribution that have more effect on the results than even a considerable approximation in the method of analysis. Concentrated loads are not applied on a line over the entire width of the arch ring, so that the correct load per unit width cannot be calculated; and, thus far, no tests have been made to determine how a concentrated load is distributed throughout the structure or what part of

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† Received by the Secretary, February 17, 1925.

the load should be assumed to be carried by the critical element for purposes of analysis. In many cases a proper assumption is the controlling factor in the design. Carefully prepared influence lines for a unit load traveling over the span are not very significant without a solution of the problem of distribution. The writer has analyzed several arches for which uniform loadings could be assumed that would agree closely with the expected traffic; but unusual concentrations produced by heavy loads that were expected to pass over the bridge occasionally controlled the design even though higher unit stresses were permitted. Various assumptions as to the lateral distribution of these concentrations, even though apparently reasonable in themselves, made considerable difference in the results. Here is a problem that must be attacked, and it is a realization of this fact that leads many engineers to the opinion that much of the refinement of mathematical processes as applied to structural design is of little value. This is certainly a wrong point of view because the truth will only be reached by the elimination of uncertainties and the sooner this can be accomplished, either by mathematical or experimental means, the better.

The next step, therefore, should be to determine experimentally how a concentrated load is distributed throughout the arch ring from the spandrel columns of an open spandrel arch or through the earth-fill and over the arch ring of a filled spandrel arch. This is a baffling problem for the designer; a correct solution is necessary in order to give any considerable degree of consistency to the design of arch bridges.

A beginning has been made by a study of the laws of distribution of concentrated loads over flat slabs, a study that throws a little light on the problem as applied to arches as well. This refers to the so-called Illinois tests* by W. A. Slater, M. Am. Soc. C. E., and to the tests by the Ohio Highway Commission. These tests, however, are insufficient and inconclusive even as they relate to flat slabs in that (1) their range was entirely too limited, and (2) they failed to consider either non-central loading or the shearing and diagonal tension stresses at the support, for loads near the support of slabs proportioned for moment in accordance with the determined law of distribution of the central load. The scope of these tests should be extended and then the investigator will be in better position to attack the problem as concerns the arch.

The author cannot receive too much praise for his paper since he has developed the first complete and satisfactory method for actually designing arch bridges as contrasted with methods of analyzing arches of assumed proportions; and there seem to be no loose ends in the development. There may be only a few engineers who have sufficient mathematical ability to follow through all his derivations (the writer does not pretend to be one of the few), but his results can be applied. The exposition and diagrams visualize the behavior of arches under load in a manner that common methods of analysis do not accomplish. His investigation of the relative economy of arches having different ratios of crown thickness to thickness at the springing line is important.

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* Referred to in *Engineering News*, July 30, 1914, and discussed by Mr. Slater in the *Journal*, Am. Concrete Inst., January, 1914.

FLOOD FLOW CHARACTERISTICS

Discussion*

By Messrs. Robert Follansbee, O. W. Hartwell, E. C. LaRue, N. C. Grover, and B. Okazaki.

Robert Follansbee,† M. Am. Soc. C. E. (by letter).‡—This paper contains the most complete tabulation of flood flows that has come to the writer's attention, and the Engineering Profession is indebted to the author for his able treatment of the subject.

The list of maximum observed discharge rates should be brought down to date where possible, especially in the case of records for stream-gauging stations still being maintained, where floods greater than those presented by the author have been recorded. Table 3 gives revised values for gauging stations in Colorado and Wyoming.

TABLE 3.—Revised Values for Maximum Floods, on Streams in Colorado and Wyoming.

Number.	Stream.	Drainage area, in square miles.	Maximum flood, in second-feet per square mile.	Date.
306	Williams Fork, Parshall, Colo.	185	14	June 14, 1918
387	Big Thompson, near Drake, Colo*	274	29	July 31, 1919
401	Bear Creek, Morrison, Colo.	180	48	July 24, 1896
557	Eagle, at Eagle, Colo†	650	10	June 3, 1914
718	Colorado (Grand), Kremmling, Colo.	2 360	9	June 9, 1912
753	Arkansas, Canon City, Colo.	3 090	6	Aug. 2, 1921
801	Colorado (Grand), Glenwood Springs, Colo.	4 560	7	June 14-15, 1918
858	Big Horn, Thermopolis, Wyo.	8 080	3.7	July 24, 1923
861	Colorado (Grand), Palisades, Colo.	8 790	6	June 16, 1921
880 (*a*)	North Platte, above Pathfinder, Wyo‡	7 410	2.5	June 26, 1917
898	Colorado (Grand), Fruita, Colo.	17 100	7	July 4, 1884

Note.—Drainage areas measured on more accurate maps.

* Drake record substituted for Loveland record.

† Eagle record substituted for Gypsum record.

‡ Record above Pathfinder substituted for record below reservoir, as former was unaffected by storage.

During 1922 and 1923, the States of Colorado and Wyoming experienced a number of severe floods, some of which were the greatest ever recorded. These are to be described in a forthcoming report of the U. S. Geological Survey.§

In addition, two other severe floods of previous years which have been recorded are not included in the author's list. Table 4 shows the maximum discharge of these floods, arranged according to the size of the drainage areas.

* This discussion (of the paper by C. S. Jarvis, M. Am. Soc. C. E., published in December, 1924, *Proceedings*) is printed in *Proceedings* in order that the views expressed may be brought before all members for further discussion.

† Dist. Engr., U. S. Geological Survey, Denver, Colo.

‡ Received by the Secretary, December 19, 1924.

§ "Some Floods in the Rocky Mountain Region", *Water Supply Paper No. 520-G*, U. S. Geological Survey.

TABLE 4.—MAXIMUM OBSERVED FLOOD DISCHARGE RATES.

Stream.	Drainage area, in square miles.	Crest discharge, in second feet per square mile.	Date.
Skyrocket Creek, at Ouray, Colo	1.0	2 000	July 20, 1923
Magpie Gulch, near Golden, Colo,	1.5	1 270	July 26, 1923
Missouri Canyon, near Masonville, Colo	2.4	1 820	June 16, 1923
Hogan's Gulch, at Eden, Colo	6.1	1 580	Aug. 7, 1904
Bayou Gulch, near Franktown, Colo	19*	460	July 28, 1922
Redstone Creek, at Masonville, Colo	21*	325	June 16, 1923
Tough Creek in Sec. 28, T. 39 N., R. 94 W., Wyo	24	62.5	July 24, 1923
Buckhorn Creek, near Masonville, Colo	40*	262	June 16, 1923
Cherry Creek, near Parker, Colo	87*	195	July 28, 1922
Paintrock Creek, near Hyattville, Wyo	164	30.2	July 24, 1923
Salt Creek, below Salt Creek Reservoir, Wyo	520	61.5	Sept. 27, 1923
Badwater Creek, at Bonneville, Wyo	794	23.4	July 24, 1923
North Fork, Shoshone, near Wapiti, Wyo	800	10.1	July 22, 1923
Powder River at Arvada, Wyo	6 050	15.7	Sept. 29, 1923
Cheyenne River, at Hot Springs, S. Dak	8 720	17.2	May 12, 1920

* Drainage area affected by storm.

O. W. HARTWELL,* ASSOC. M. AM. SOC. C. E. (by letter).†—The author has made a valuable contribution to the long list of papers on this subject. In the past much has been written about flood flows, and many formulas have been evolved in attempts to predict the size of floods. This problem will be studied again and again in the future, with results increasing in value as the fund of actual flood data increases.

The author has overlooked one source of information which will give him many valuable additions to Table 2.‡ In all the recent *Water Supply Papers* of the U. S. Geological Survey the maximum and minimum discharge during the total period of record is published for each gauging station. In the report for any given year may also be found the extremes of discharge for that year.

The author dismisses consideration of the time element too hastily. There are many projects where human life is not endangered and where the cost of various degrees of protection to property must be compared with corresponding losses. Such problems can best be solved by the theory of probability. In this connection, Allen Hazen, M. Am. Soc. C. E., wrote,§ in 1914:

"As time goes on, data covering longer periods and more streams may change some of the numerical values; but the underlying idea of treating the recurrence of floods as a matter of probabilities, to be determined by an examination of the records of many streams, will stand."

Ten years have already been added to the continuous records for a large number of the stream-gauging stations which existed in 1914, many new stations having been installed since that time. Furthermore, the accuracy of flood data has been greatly increased through the development of the automatic water-stage recorder and through other improvements in methods of conducting stream-gauging work. It will take 1 000 years to obtain an actual 1 000-year record, but long before the end of this period engineers will be able to determine the 1 000-year probability within a reasonable degree of accuracy.

* District Engr., U. S. Geological Survey, Trenton, N. J.

† Received by the Secretary, December 30, 1924.

‡ *Proceedings*, Am. Soc. C. E., December, 1924, p. 1563.

§ *Transactions*, Am. Soc. C. E., Vol. LXXVII (1914), p. 626.

E. C. LaRue,* M. Am. Soc. C. E. (by letter).†—Since the automobile became a dependable means for long-distance travel the demand for good roads has been almost universal. During recent years hundreds of millions of dollars have been expended for the construction of automobile highways in the United States. The cost per mile of many of the hard-surface highways is comparable with that of railroad construction. It is generally conceded that some of this road work has been done without providing adequate structures to take care of transverse drainage. How many engineers make a serious study to determine the magnitude of the floods which may be expected in each running stream or dry wash?

There is much to be said in defense of the engineer. He is not always responsible for the poor location of a railroad or highway, or for drainage structures of inadequate capacity. The author states that "an important Western railroad company located nearly 100 miles of line along a valley floor, safely above the reach of the maximum assumed floods, only to experience its destruction within a few years". Some years ago, the writer talked with the engineer who had charge of the location of this railroad. He had strongly recommended another location to avoid the floods which he, at that time, knew to be dangerous, but was overruled by those who were financing the construction. The estimated initial cost for the location in the valley was less than that recommended by the chief engineer.

The author has presented a simplified formula‡ for computing maximum floods which may be expected from a given drainage basin. Although this is an important step in the right direction, he has left plenty of work for the engineer who must design the structures, and who must make certain that they will carry the maximum load with safety, that they will not be damaged by the maximum expected floods, and that their estimated costs are reasonable. It will be noted that the author's formula $\left(Q = R\sqrt{M}\right)$, carries a factor which may vary from 100 to 10 000. This formula will unquestionably prove useful in actual practice. However, the engineer must make extensive studies to determine the flood flow characteristics of the streams in the territory through which he must build a railroad or highway. Where the construction of a dam is being considered the studies of flood flows may be confined to its particular drainage basin.

The writer, being familiar with the flood flow characteristics of the Colorado River and many of its tributaries, has applied the author's formulas for the purpose of comparing the calculated values for flood flows with those obtained from recorded data or traditional evidence. It was found that the formulas give erroneous results when used to determine flood flow maxima on the main stream of the Colorado River. According to the formulas the greater the area of drainage basin, the greater the maximum flood. This is not true of the Colorado River. The maximum flood flow at Lees Ferry (drainage area, 108 000 sq. miles) may be 250 000 sec-ft., while the maximum

* Hydr. Engr., U. S. Geological Survey, Pasadena, Calif.

† Received by the Secretary, February 7, 1925.

‡ *Proceedings*, Am. Soc. C. E., December, 1924, Papers and Discussions, p. 1554.

flood at the Laguna Dam (drainage area, 186 000 sq. miles) may be only 200 000 sec-ft. The records show that on this stream the maximum flood flows occur at a point 800 to 850 miles above its mouth (except floods from the Gila River).

The writer believes that the maximum flood that may be expected in a stream with a drainage basin of 1 000 sq. miles, or more, can be estimated with greater accuracy than in streams having drainage basins of a few square miles. The greatest run-off per square mile of drainage area occurs in streams having small drainage basins; however, the frequency of these floods is much less than those which occur in streams having large drainage basins. The engineer has to decide as to how much money may reasonably be spent on structures where the maximum flood may not occur once in 100 or 200 years. It is true that the 100-year flood may occur next year; it is also true that if the structures over every dry wash were built to withstand the 100 or 200-year maximum flood, the mileage of highways to be constructed would be seriously reduced. Where the problem does not involve possible loss of human life, it seems reasonable for the engineer to design flood by-pass structures on a less conservative basis, taking into account the cost of replacing structures which may be destroyed once in 50 years.

The author's graphical tabulation of run-off records (Plate VIII*) is a valuable contribution to the subject, and he is to be commended for his exhaustive search for the data which form the basis for this diagram. Its accuracy and value are only limited by the flood-flow data which were available. The diagram, perhaps, fails to show the maximum 50-year or 100-year flood for many of the streams due to the lack of available records, but the author has plotted sufficient data to show in a general way the limits of maximum floods that may be expected in streams with drainage basins varying in size from less than 1 sq. mile to more than 1 000 000 sq. miles. The diagram would obviously be more valuable if sufficient data were available to show the maximum flood that has occurred during the past 100 years on each of the respective streams considered.

The author is unfair to himself, when in his conclusion he says:

"In this paper, the writer does not assume to have solved any particular problem, but merely lists certain data and considerations that are often left unrelated or neglected entirely."

He has suggested a simple formula which may prove to be of considerable value to engineers who must calculate maximum flood flows that may be expected in a given region. The paper as a whole may prove an important factor in furthering the work that should be done to determine flood dangers along streams throughout the country.

N. C. Grover,† M. Am. Soc. C. E. (by letter).‡—This paper emphasizes the importance of a thorough study of floods, the need for which was anticipated by the Board of Direction of the Society when, in 1922, it created a Special

* *Proceedings*, Am. Soc. C. E., December, 1924, Papers and Discussions, p. 1555.

† Chf. Hydr. Engr., U. S. Geological Survey, Washington, D. C.

‡ Received by the Secretary, February 13, 1925.

Committee on Flood Protection Data. That Committee, after compiling the records of many floods, has recommended that the Society undertake an elaborate study of flood data and the publication of the results in such form as to afford engineers a satisfactory working basis for handling the many problems of design and construction related to floods.

In the absence of sufficient data to serve as a basis for studying floods by river basins, or even by regions, the author has followed the common practice of showing on a single diagram (Plate VIII*), the available flood data for a wide area (in this instance, the whole world) and of plotting thereon limiting, average, and regional curves of various kinds that may serve to aid him and others in establishing elevations of highways and railroads and capacities of culverts and other bridge openings and of spillways and flood channels. The diagram has served to generalize the available data and to show how the regional curves are related to all the data and to the limiting curves of the world's floods. He has shown also, by sixteen symbols, the flood data for different regions, but the plotting of so many points on a single diagram has made difficult the separate utilization of the information pertaining to any particular region. He has also presented a diagram (Fig. 6*) showing the various flood formulas and, by modifying one of these formulas, has suggested a possible method of approach to the problems that arise in certain fields of engineering.

The results are evidently not wholly satisfactory to the author, as he "does not assume to solve any particular problem but merely lists certain data and considerations that are often left unrelated or neglected entirely." His inability to reach satisfactory conclusions should perhaps have been anticipated, as the only data available are unsystematic and even incomparable, because they include without discrimination both peak discharges and averages for days of maximum discharge. As a net result of reading the paper one is left with the feeling that organized engineers, and specifically the Society, should find a better way to serve the profession by providing a rational and adequate basis for treating the many important problems related to floods. Evidently, the author is similarly impressed, as he states in his "Conclusion"† that his purpose in preparing this paper will have been accomplished when "a definite campaign to determine flood dangers and the advisable measures for control" has been inaugurated.

Floods are essentially local phenomena in which precipitation is the major factor, even though temperature, in its effect in storing water as snow and ice and, later, in releasing it by melting, may at times be controlling. Other factors also serve to modify the effects of precipitation in varying degrees. Precipitation and temperature vary widely within relatively short distances, so that each of the factors modifying the effect of precipitation in producing floods is strictly local in its application. Floods, therefore, must be studied locally by river systems and sometimes even by tributaries, as well as broadly by continents, in order to reach reliable conclusions. A composite picture

* *Proceedings*, Am. Soc. C. E., December, 1924, Papers and Discussions, p. 1555.

† *Loc. cit.*, p. 1561.

of the world's floods will disclose, of course, the limits of floods that are due to the worst conditions of rainfall and run-off, but will not show what may reasonably be expected on any particular stream. Obviously, engineers should not design structures for all rivers on the basis of floods that occur in only a few. The cost would be unwarranted because unnecessary, even though such structures would have the saving grace that they would surely be ample and, therefore, never the causes of physical disaster. Moreover, much legitimate development would be prevented because of prohibitive costs.

California has made a study of the floods on the rivers of that State and has published the results in graphic form.* Similar studies should be made for many rivers in other States by using as the principal source of information the systematic records of discharge collected by the U. S. Geological Survey and supplementing them by all other pertinent data obtainable. This is the first step in the program of the Special Committee on Flood Protection Data, for which it is now seeking sufficient funds to warrant the employment of a mature and competent engineer who can work continuously under the direction of the Committee. Following this step the Committee expects, if available funds permit, so to correlate floods with climate, topography, geology, vegetation, and other modifying factors that reasonably accurate predictions of floods may be made not only for the rivers in the United States that have been systematically gauged, but also for the many others for which actual records of discharge are not available.

The writer feels, therefore, that although this paper has direct value in meeting the present needs of engineers for aid in applying the meager available flood data to immediate problems, its greatest value is indirect, for it accentuates the range and importance of flood problems and the unsatisfactory data and methods now available to engineers for solving them. He hopes that its presentation and discussion will assist in promoting a rational study of the magnitude and control of floods in the rivers of the United States and the publication of the results in form for ready use, thereby making available to engineers not only the mass of flood data that is awaiting compilation and analysis, but also methods for its application to particular problems. This will change the present unsatisfactory, expensive, and even dangerous procedure into one that will serve as a basis for economic and safe development and that will bring credit to the profession.

B. Okazaki,† M. Am. Soc. C. E. (by letter).‡—It is well known that where an improved river, in its primitive state, overflows the natural river banks and adjoining lands, the actual maximum discharge at a point down stream from the valley subject to overflow is far less than if the river were diked or "short-cut" to prevent inundation. The natural controlling action caused by the temporary reserve or detention of flood water over the submerged area makes it sometimes necessary to fix the criterion for maximum flood discharge in determining the stability or size of a structure contemplated and the extent

* "Flow in California Streams," *Bulletin No. 5*, Department of Public Works, California.

† Engr., Upper Liao River Conservancy Board, Newchwang, China.

‡ Received by the Secretary, March 2, 1925.

of waterway to be provided for probable future floods. The known maximum flood discharge may be greatly exceeded if the river is improved by short-cuts or embankments that do away with the natural flood reserve.

In dealing with a river in its natural state, the writer suggests the need for considering this controlling action due to overflow. As the volume of water temporarily impounded over a large area changes continually, it is difficult to determine the overflow in reserve and the time of maximum intensity, and, likewise, the quantity to be added to the ordinary maximum discharge in order to forecast the probable future discharge after the up-stream portion has been improved.

The writer made a careful investigation of the Ishikari River, which is one of the two largest in Japan and which had long been unregulated. It was found that the known maximum discharge, in July, 1904 (150 000 cu. ft. per sec.) at a place called Tsuishikari, which is about 25 miles up stream from the river mouth and which has 5 600 sq. miles of tributary drainage area, should be doubled, namely, 300 000 cu. ft. per sec., and that the observed run-off of 26.8 cu. ft. per sec. per sq. mile might be increased to 53.6 cu. ft. Therefore, this information should be considered in plans for improvement.

It should also be noted that on the same river for a point called Kamuikotan about 132 miles up stream from the mouth where the drainage area is 1 400 sq. miles and the impounding action is negligible, the same flood gave a maximum discharge of 129 000 cu. ft. per sec., or 92.2 cu. ft. per sec. per sq. mile.

DEVELOPMENT OF HIGHWAY TRAFFIC IN CALIFORNIA

Discussion*

By Messrs. Arthur E. Loder, Watt L. Moreland, and T. E. Stanton, Jr.

Arthur E. Loder,† Esq.—The development of the highway traffic of California, as presented by Dr. Hewes,‡ cannot fail to impress every one with the seriousness of the highway transport problem. He shows that within a period of two years the average summer travel on the State Highway System has increased 47.4 per cent.

The combined registration of automobiles and trucks was 574 623 for 1920, in which year the first traffic census was taken, and 861 704 for 1922, during which the second census was taken. This increase of 50% in registration in the two-year period is slightly in excess of, and shows cause for, the increase of 47.4% in the average summer traffic found in the traffic survey. The increase in traffic shown during the two-year period under study has undoubtedly been exceeded by the two-year period, 1921 to 1923, during which motor-vehicle registration in California increased 61.6 per cent.

The average summer traffic may be expected to increase in approximately the same ratio as motor-vehicle registration. Both factors indicate the potential motor traffic. With generally improved roads the maximum traffic must be expected to come out in the highways in summer (and, in fact, on any occasion during the year) and to demand a reasonable degree of comfort and efficiency in the use of the highway.

It is the peak load of traffic that gives rise to most of the difficulty. This exceptional traffic comes on Sundays or holidays in answer to the demand for recreation. It is highly desirable that it be accommodated, in so far as practicable, but the matter of first concern, from an economic standpoint, is the giving of efficient highway service to the largest possible total volume of traffic, and this consideration will probably make it impossible to provide highway facilities of sufficient width and extent to handle all maximum traffic.

It is probable that some form of regulation may be found, which will have a tendency to smooth down the peaks in the traffic profile. In fact, present conditions are such that a large percentage of city motorists must remain at home on Sundays because of the limited highways. Although highway facilities appear to be good in the vicinity of centers of population, they are

* This discussion (of the paper by L. I. Hewes, M. Am. Soc. C. E., presented at the meeting of the Highway Division, Pasadena, Calif., June 19, 1924, and published in March, 1925, *Proceedings*), is printed in *Proceedings* in order that the views expressed may be brought before all members for further discussion.

† Chf. Engr., California State Automobile Assoc., San Francisco, Calif.

‡ *Proceedings*, Am. Soc. C. E., March, 1925, Papers and Discussions, p. 377.

found to be actually very inadequate when compared with the large number of motor vehicles, which, in California, has reached the high mark of 1 to each 3.5 persons. These conditions, and the fact that city motorists want to do their driving in the country, indicate one of the principal reasons why city motorists are contributing financially to the major highway improvement in the country.

Although California has been the scene of an unusually rapid increase in the ownership and use of motor vehicles, this is a problem which is confronting all the States, and especially the densely populated centers, as shown by the fact that the registration of motor vehicles in the United States for 1923 showed an increase of 23.6% over the previous year.

There is no occasion for alarm or sensational attempts at guessing the answer. The fundamental laws of economics will regulate abnormal tendencies for the best ultimate good of the country and will finally make for harmony between the urge for motor transport and the ability of the public to pay the bill. In the final analysis, this will serve as a check on the development of the motor industry and of motor transport to prevent it from growing too far in excess of possible highway facilities or too far beyond the ability of man to supply himself with the remaining necessities of life. There is much comfort in the knowledge that, although man may closely pursue his winged imagination in its search for unlimited individual transportation, he must always remain sufficiently close to earth and to common sense to satisfy his instinctive demands for food and shelter, the preservation of the race, and the esteem of his fellow-man.

There is only a remote probability of reaching the so-called "saturation point" in ownership and operation of motor vehicles. The increase will continue much as in the past with occasional interruption or slowing down as financial periods swing from expansion to depression. The ability of the public to finance highways will likewise continue to increase but, as at present, it will always be found lagging several laps behind the demand. Obviously, the future will see such large funds going into the maintenance, reconstruction, and new building of highways, as to make present budgets appear as small beginnings.

If increase in the number of motor vehicles, combined with increase in population and highway facilities continues, as in the past few years, a registration of 2 500 000 motor vehicles may be expected in California by 1930. This would indicate a total traffic four times as great as that measured in 1922. To this will be added a material increase in traffic caused by motor guests. Approximately 28 000 cars from other States and countries toured in California during 1922 and more than 66 000 during 1923, or an increase of 135% in one year. If this condition existed when the interstate connections were still unimproved, what may be expected in the next few years in the way of motor guests, when a number of transcontinental highways have been improved to such an extent as to make the trip a pleasure instead of an adventure?

These considerations of highway traffic point to the necessity for immediate and continued activity on the completion of the State Highway System and the

widening or duplication of a number of the most important inter-city and suburban routes.

During the present year (1924), the California State Highway Committee of Nine is making a study of the status of the State Highway System from an engineering and economic standpoint and is required to submit recommendations to the Governor and the next Legislature on such questions as State highway classification, desirable progress and policies affecting order of construction, and a method of taxation which may equitably place the burden of cost and provide funds for continuing the work toward completion. These studies of the development of highway traffic point to the fact that the State will probably never find itself able to say that its highway system is completed. The highways must continue to grow in length, width, and thickness, as long as traffic continues to increase as it has in the past.

While it may be said that the State at the present moment faces the question of providing means of raising approximately $200 000 000 for the completion of work which seems desirable on the State Highway System, to provide for present traffic conditions, it is probable that even though this work may be performed within the next ten years the State would then find itself facing an even larger financial problem to care for the volume of traffic which may then exist. The probable increase in traffic points to the desirability of now placing the State's highway program on a "pay-as-you-go" basis to prevent paying at the rate of $2.08 (principal and interest) for each dollar available for construction, as has been necessary—and justifiable—on the first $73 000 000 of bond money which has already gone into the work.

The seriousness of the highway traffic problem of the State, and the financial responsibility which it brings, suggests the desirability of considering ways and means by which the usefulness and efficiency of inter-city highways can be increased. Anything that will have a tendency to increase the number of vehicles which can be given efficient highway service over the road, will help toward solving the problem and will have a tendency ultimately to conserve funds. Some highways now handle a large total count of traffic on busy days, but when people are required to proceed so slowly and stop so frequently because of traffic jams at the frequent intersections and through each village, they soon lose all interest in the trip and take the first detour home or spend many aggravating hours on what should be a short trip. In such cases it is true that a large total number of vehicles pass over the road, but each driver will tell you that he receives only about 10% efficiency as far as highway service is concerned.

Highway service should be regarded as the number of vehicles passing over the road modified and reduced by the efficiency of service. Some day this factor will be given far more consideration than at present. This will ultimately bring about the location of inter-city highways in such manner as to pass by the side of intervening towns and villages instead of through their congested streets.

The speed of traffic and, therefore, the efficiency of highway service must be increased by reducing to a minimum the number of highway intersections and

road crossings at grade and providing the smallest possible number of intersecting avenues leading into the adjacent towns or villages. The future inter-city highways will be located and built so as to be safe for continuous speeds of 35 to 40 miles per hour, and probably more, operating under such conditions that vehicles can follow each other more closely than would be safe on the present highways with their frequent intersections and interruptions of the stream of traffic. The future great highways between the largest centers of population will probably be laid out with a view to these considerations under new legislation yet to be devised, whereby the highways can be under the complete ownership and control of the State instead of being merely an easement over property which now exercises the right of creating as many dangerous intersections and frontage obstructions as owners and land "butchers" can devise.

Under present conditions, an important suburban or inter-city highway is no sooner paved to an ample width than sub-divisions are laid out and store fronts erected. This is followed by an unlimited number of street intersections and continuous parking of vehicles. To make the situation worse, school officials, perhaps through misguided civic pride, frequently locate beautiful new school buildings directly on main traffic thoroughfares, thus subjecting school children to the greatest possible danger to life and limb. Primitive races may yet be found where youth is subjected systematically to all manner of physical hardship and danger with a view to making him agile in dodging dangers and useful in war, but in this civilization it is doubtful if there is any justification for such a system of training, and the more thoughtful communities will place their schools in quiet places.

The future express highways between large cities will probably consist of one or more high-speed motorways, fenced or walled on either side, except at intervals of perhaps 5 or 10 miles where turn-outs and inlets from intersecting avenues will be provided. The high-speed motorways will be located on land owned in fee by the State and safeguarded under future legislation so that it may perform its function of serving the greatest possible number of people in the most efficient manner. On both sides the high-speed roadway will be flanked with narrower but continuous local roads to which the abutting property will have free and unhindered access as at present. At the limited entrances to the high-speed motorways there will be located grade separations and wide turn-outs and inlets arranged so as to avoid the necessity for left-hand turns or crossings at grade on the express highway. Grade separations will be provided wherever transverse roads must cross. With such a layout local traffic will be fully protected from high-speed traffic and the high-speed traffic will not be interfered with by local and slow traffic. Local traffic will follow the parallel local roadways until reaching the first intersection where it may, if it desires, enter the high-speed motorway.

It is true that at present the foregoing layout would be necessary in only a few parts of the country, but it is believed the time will come when substantially such a plan will be in use. If motor transport is to continue its development along with inter-city through automobile traffic, there will come finally an irresistible demand for such a plan.

Watt L. Moreland,* Esq.—This excellent paper brings out a number of interesting facts. As a truck manufacturer, the speaker is particularly interested in the statements that 60% of the trucks are overloaded, that the average length of hauls is 31 miles (which is about twice the length of hauls in the Eastern States), that the number of persons per automobile in California is a little less than 3.5, and that the average speed of passenger busses is 33.1 miles per hour.

This overloading of trucks is a constantly increasing problem for the truck distributor due to the limiting of weights and capacities by State authorities and to the constantly increasing tendency to buy lighter trucks and overload them. While the figures show that although 60% of trucks are overloaded as judged by their rated capacities, only 11½% were overloaded according to tire capacities. Of course, highway engineers are not particularly interested in the capacity of motor trucks to take overloading and the consequent high maintenance costs and short lives. What particularly appeals to engineers is the overloading of the maintenance charges of highways. It is doubtful whether any practical way can be found to prevent overloading beyond the manufacturer's rating. There is such a difference in rated capacities according to the weight of the truck and the grades of steel, etc. entering into the construction, that it would seem to preclude any such limiting legislation.

The long mileage, the large number of cars, the high speeds maintained by busses in California are easily explained, of course, by the miles of magnificent scenery, the splendid highways passing through non-congested areas, and the rapid growth of suburban and country districts. The extension of these highways into the mountains and to the seashore makes it imperative that if people are to enjoy the beauties of Nature and to secure the full advantages of the highways, they must own automobiles. A citizen of California certainly lives a better life—a more healthful life—and gets a greater enjoyment because of the highways and passenger cars than citizens in other States who do not enjoy these advantages in the same degree.

The growth of Los Angeles, or Southern California, for that matter, has kept pace with the automobile and one of the reasons for this growth has been the transportation facilities afforded by the motor vehicle.

The absence of tenement houses in Los Angeles can be credited to the passenger car, which makes it possible to live in the suburban districts and still do business within the city. Nearly all the working people now have light cars in which they travel to and from work.

The great area of the State, with its comparatively limited railroad mileage, has offered, and does offer, a fertile field for the development of motor-vehicle transportation lines, both freight and passenger. The railroads are now coming to realize the advantages to them of motor vehicles as feeders; in fact, railroads all over the country are eliminating short-line roads and turning over this work to the motor vehicle.

One great advantage that comes to the owner of an automobile has been the lower maintenance cost due to the splendid hard-surfaced highways.

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Greater efficiency in transportation means a reduction in cost of living. Highways should not become the football of politics, but should be managed for the benefit of all the people.

Although the increase of traffic in the past has been phenomenal, it will be much greater in the future; undoubtedly, the bulk of all future transportation will be by motor vehicles. With the practical elimination of the local steam train, the electric lines, in both city and suburban use, are now dividing the traffic with motor trucks. The speaker expects to see electric lines and railroad lines utilizing State highway systems and motor-vehicle transportation to the utmost.

The State highway system must be enlarged. California was one of the first, if not the first, State in the Union to build highways; a great deal had to be learned. These first highways were not properly constructed for the traffic, but since that time there has been a constant improvement. As the traffic increases the width must be increased and a careful study of traffic made to determine the advisability of one-way main arteries as compared with extra wide highways; the laws will have to be modified perhaps to permit of passing heavy loaded vehicles on the right side instead of only on the left, as at present. A great deal of attention should be given to the movement of traffic on the roads, and ways and means should be devised for insuring a steady flow. The benefit from good highways is shared by every citizen.

It would seem that maintenance costs have been excessive. This was brought about largely by inadequate highways in the first place—the surface and thickness were not of the correct dimensions and not enough attention was paid to limiting the loads carried. Imperfect highways, overloaded—two features that are now largely corrected—undoubtedly are responsible for this high maintenance cost; on the newer highways, with the regulation of loads, this cost should rapidly decrease.

Because of the many factors involved it is difficult to determine the economical load. It would seem that the present weight limit, fixed by the capacity of the highways to carry the load, is proper, and that as the highways are strengthened and special road-beds built, this limit should be increased.

Motor vehicle manufacturers are endeavoring to do their part by better construction of their products, using better material, better distribution of metal, and better distribution of load. Two or three manufacturers are announcing six-wheel vehicles, a feature which enables them to carry added loads under the law. As time goes on, this will increase until the economic capacity of different vehicles is definitely determined.

One of the greatest needs is the elimination of obstructions, such as grades, hills, narrow roads and crossings, and roads through a valley which acts like a bottle neck. All these should be removed and traffic allowed to flow smoothly.

The immense areas that will be brought into cultivation due to the many Western reclamation projects, will undoubtedly require more highways; the development of the back countries and the agricultural districts means more highways. History teaches that the better the transportation facilities, the higher the civilization.

T. E. STANTON, JR.,* M. AM. SOC. C. E.—The development of motor-vehicle traffic in the United States opens an interesting and broad field for study to the highway economist.

Californians have been prone to consider, as Dr. Hewes shows,† that the position of California with respect to highway transportation is unique in that "the very large motor vehicle registration, the outdoor climate, and the well developed highway system of the State combine to make highway traffic exceptional." It is interesting to note, however, the extent to which the development in California during the last ten years has been paralleled by other States.

Motor-Vehicle Registration.—It is true, as published statistics show, that there are more motor vehicles in California in proportion to the population than in any other State. A further study of these statistics, however, apparently demonstrates that it is only in this one respect that California occupies a unique position or one at variance with the average conditions found throughout the Union.

Two very interesting tables have been published by the U. S. Bureau of Public Roads.‡ An analysis of the figures in these tables—one a summary of combined passenger car and motor truck registrations for the years 1913 to 1923, and the other the gross receipts from gasoline taxes during 1923 in the States levying such a tax—shows that California, instead of occupying a position either at one extreme or the other, lies remarkably close to the average development throughout the country during the last decade.

The development of highway traffic in California during this decade has been remarkable, but so also has been the development throughout the country as a whole. Whether this development has resulted from improved highways, or whether the improvement in methods of transportation has been the occasion of the demand for more and better highways, may be a debatable question, but there is evidence that these two impelling motives have gone hand in hand, each spurring the other to greater efforts.

Referring again to the statistics mentioned,* it appears that in 1913 the combined passenger car and motor truck registration in all the States amounted to 1 258 062 and that ten years later, in 1923, the registration was 15 092 177, an increase of 1 100 per cent. In the same period, the registrations in California increased from 100 000 to 1 100 283, or an even 1 000 per cent.

The smallest percentage of increase was in North Dakota, with 620%, and the greatest in Oklahoma, with 10 120%, but even with this large percentage increase Oklahoma had a total registration in 1923 of only 307 000, as compared with 1 100 283 in California. It will thus be noted that, as far as percentage is concerned, the increase in California does not present any abnormal elements that do not apply with equal force throughout the country as a whole. Rather is the fact emphasized that the development of highway traffic has been and is a major problem confronting the whole country.

* Asst. State Highway Engr., Sacramento, Calif.
† *Proceedings*, Am. Soc. C. E., March, 1925, Papers and Discussions, p. 377.
‡ *Public Roads*, April, 1924, pp. 16–17.

a problem which offers the same difficulty of solution in other States as in California.

In the light of these statements, the two traffic census studies conducted by the U. S. Bureau of Public Roads in California in 1920, 1922, and 1923, as well as those now being carried on by the California State Highway organization, afford statistical data of direct interest and value to other States and thus doubly justify the expenditure of Government funds for the purpose. In Table 4 will be found figures showing the percentage increase in each State.

TABLE 4.—SHOWING PERCENTAGE INCREASE IN MOTOR-VEHICLE REGISTRATION—ALL STATES DURING TEN YEARS FROM 1913 TO 1923.*

State.	Pleasure Car and Truck Registration.			Percentage of Increase.		
	1913.	1918.	1923.	1913-18.	1918-23.	1913-23.
Alabama	5 300	46 171	126 642	771	175	2 290
Arizona	3 613	23 905	49 175	562	106	1 261
Arkansas	3 583	41 458	113 300	1 058	173	3 060
California	100 000	407 761	1 100 283	308	170	1 000
Colorado	13 000	83 244	188 956	540	127	1 351
Connecticut	23 200	86 067	181 748	271	111	684
Delaware	2 440	12 955	29 977	431	131	1 129
Dist. of Columbia	4 000	30 490	74 811	663	145	1 770
Florida	3 000	54 186	151 990	1 706	181	4 966
Georgia	20 000	104 676	173 889	423	66	769
Idaho	2 113	32 289	62 379	1 427	93	2 852
Illinois	94 656	389 620	969 331	311	149	924
Indiana	45 000	227 160	583 342	405	157	1 196
Iowa	70 299	278 313	571 061	296	105	712
Kansas	34 550	189 163	375 594	448	99	987
Kentucky	7 210	65 884	198 377	814	201	2 651
Louisiana	10 000	40 000	136 622	300	242	1 266
Maine	11 022	44 572	108 609	304	144	885
Maryland	14 217	74 666	169 351	425	127	1 090
Massachusetts	62 660	193 497	481 150	209	149	667
Michigan	54 366	262 125	730 658	382	178	1 244
Minnesota	46 000	204 458	448 187	344	119	875
Mississippi	3 850	48 400	104 286	1 157	115	2 608
Missouri	38 140	188 040	476 598	393	153	1 152
Montana	5 916	51 053	73 828	762	45	1 148
Nebraska	13 411	173 374	286 053	1 191	65	2 030
Nevada	1 091	8 159	15 699	648	92	1 340
New Hampshire	8 237	24 817	59 604	202	140	623
New Jersey	51 360	155 519	430 958	203	177	739
New Mexico	1 898	17 647	32 032	830	81	1 586
New York	134 495	459 288	1 204 213	242	162	795
North Carolina	10 000	72 313	246 812	623	241	2 368
North Dakota	15 187	71 678	109 266	371	52	620
Ohio	86 156	412 775	1 069 100	379	159	1 140
Oklahoma	3 000	121 500	307 000	3 950	153	10 133
Oregon	13 975	63 324	165 962	353	162	1 088
Pennsylvania	80 178	394 186	1 043 770	391	165	1 202
Rhode Island	10 295	36 218	76 312	252	111	642
South Carolina	10 000	55 492	127 467	455	129	1 175
South Dakota	14 457	90 521	131 700	526	45	811
Tennessee	10 000	63 000	173 365	530	175	1 634
Texas	32 000	251 118	688 233	685	174	2 050
Utah	4 000	32 273	59 525	707	84	1 390
Vermont	5 913	22 553	52 776	281	135	790
Virginia	9 022	72 228	218 896	701	203	2 326
Washington	24 178	117 278	258 264	386	120	970
West Virginia	5 144	38 750	157 924	654	308	2 970
Wisconsin	34 346	196 253	457 271	472	133	1 233
Wyoming	1 584	16 200	39 881	924	146	2 415
Total	1 258 062	6 146 617	15 092 177	389	146	1 100

* Based on Table II, p. 16, Vol. 5. No. 2, *Public Roads*, April, 1924.

The relatively small number of motor vehicles per square mile in California, however, does offer greater problems in financing than in the more populous States of smaller area where the bulk of the traffic can be served with less mileage of main arterial highways.

Gasoline Tax.—A study of the revenue per motor vehicle derived from the gasoline tax offers another interesting line of investigation. The 1923 session of the California Legislature passed a Gasoline Tax Bill calling for a tax of 2 cents per gal. The tax went into effect on September 30, 1923, so that the income for only one-quarter of the year is available for purposes of comparison.

In estimating the probable revenue which would accrue from this method of taxation it has been customary to assume that because of the mildness of the climate in the populous sections of California the year round traffic would average somewhat higher than in States with a more rigorous winter climate and that the corresponding revenue per car for each cent of gas tax would be somewhat greater.

As will be seen from Table 5, such an assumption was not well founded. It is true that the revenue for only one-quarter of the year, in the late fall and early winter, is available for direct comparison; however, the receipts for the first quarter of 1924 have been little in excess of the preceding quarter, thus indicating that, although the spring and summer months will undoubtedly show an increase, the average for the year will not be materially in excess of that shown in Table 5 for the last quarter of 1923.

For purposes of comparison it has been necessary to determine the probable revenue in each State in one year for each cent of tax. This has been difficult owing to the fact that the tax was in effect in a majority of the States only part of the year or that a change in rate was put into effect during the year.

The average revenue per car per State is found to be \$4.46 for each 1 cent of tax. The corresponding revenue per car in California for the last quarter of 1923 was \$4.58.

Several States, notably Texas with a tax of \$3.26 per car, show an unexplainedly low return, whereas other States with a rigorous winter climate give remarkably high returns, notably New Hampshire, the highest of all, with \$6.56 per car, and Maine, with \$5.45 per car. The four Southern States of Alabama, Florida, Georgia, and Louisiana, show a high average return of \$5.41 per car. Whether it is the poor condition of the highways in these States resulting in a less mileage per gallon of gasoline or some other unsuggested cause or causes is for the investigator of the future to determine. Without doubt the variation in revenue per car in the different States offers a decidedly interesting field for study.

Construction and Maintenance Problems.—The rapid development of traffic in California presents many serious problems, not the least of which are those of design and financing. These two problems are, in fact, the major ones and to a very great extent must be considered together.

When State highway construction was started in California in 1912, there were approximately 91 000 registered motor vehicles; a year later, there were only a few more than 100 000.

TABLE 5.—GASOLINE TAXES, 1923.*

State.	PERIOD, 1923.				PERIOD, 1923.				Rate, January 1, 1924, in cents.	Gross receipts, 1923.	Estimated receipts in one year for each cent gas tax, based on 1923 gross receipts.	Registration, 1923.	Revenue per car.
	From	To	Months.	Rate, in cents.	From	To	Month	Rate, in cents.					
Alabama	.....	.....	.....	....	Mar. 1	Dec. 31	10	2	2	$1 133 085.49	$679 851.29	126 642	$5.37
Arizona	Jan. 1	June 9	5.3	1	June 9	Dec. 31	6.7	3	3	474 123.04	224 000.00	49 175	4.35
Arkansas	Jan. 1	Apr. 1	3	1	Apr. 1	Dec. 31	9	3	4	1 219 198.75	487 679.50	113 300	4.30
California	.....	.....	0	0	Oct. 1	Dec. 31	3	2	2	2 518 893.00	5 037 786.00	1 100 283	4.58
Colorado	Jan. 1	Aug. 1	7	1	Aug. 1	Dec. 31	5	2	2	846 353.12	597 425.73	188 956	3.16
Connecticut	.....	.....	.....	....	Jan. 1	Dec. 31	12	1	1	880 222.70	880 222.70	181 748	4.84
Delaware	.....	.....	.....	....	Apr. 22	Dec. 31	8.3	1	2	88 579.28	128 066.48	29 977	4.27
Florida	Jan. 1	July 1	6	1	July 1	Dec. 31	6	3	3	1 641 042.25	820 521.12	151 990	5.40
Georgia	Jan. 1	Oct. 1	9	1	Oct. 1	Dec. 31	3	3	3	1 502 503.49	1 001 668.99	173 889	5.76
Idaho	.....	.....	.....	....	Apr. 1	Dec. 31	9	2	2	396 487.18	264 324.78	62 379	4.24
Indiana	.....	.....	.....	....	June 1	Dec. 31	7	2	2	2 906 428.25	2 491 224.21	583 342	4.27
Kentucky	.....	.....	.....	....	Jan. 1	Dec. 31	12	1	1	680 435.30	680 435.30	198 377	3.43
Louisiana	.....	.....	.....	....	Jan. 1	Dec. 31	12	1	1	754 437.85	754 437.85	136 622	5.52
Maine	.....	.....	.....	....	July 7	Dec. 31	5.8	1	1	286 076.97	591 883.39	108 609	5.45
Maryland	.....	.....	.....	....	Jan. 1	Dec. 31	12	1	2	688 304.02	688 304.02	169 351	4.06
Mississippi	.....	.....	.....	....	Jan. 1	Dec. 31	12	1	1	467 855.53	467 855.53	104 286	4.49
Montana	Jan. 1	July 1	6	1	July 1	Dec. 31	6	2	2	441 249.10	294 166.06	73 828	3.98
Nevada	.....	.....	.....	....	Mar. 20	Dec. 31	9.3	2	2	115 843.24	74 737.57	15 699	4.76
New Hampshire	.....	.....	.....	....	July 1	Dec. 1	5	1	2	163 064.64	391 375.13	59 604	6.56
New Mexico	.....	.....	.....	....	Jan. 1	Dec. 31	12	1	1	165 000.00	165 000.00	32 032	5.15
North Carolina	Jan. 1	Apr. 1	3	1	Apr. 1	Dec. 31	9	3	3	2 909 904.74	1 163 961.90	246 812	4.72
North Dakota	.....	.....	.....	....	Jan. 1	Dec. 31	12	1	1	461 081.71	461 081.71	109 266	4.22
Oklahoma	.....	.....	.....	....	July 1	Dec. 31	6	1	1	599 000.00	1 198 000.00	307 000	3.90
Oregon	Jan. 1	May 24	4.8	2	May 24	Dec. 31	7.2	3	3	1 958 141.37	753 131.30	165 962	4.52
Pennsylvania	Jan. 1	July 1	6	1	July 1	Dec. 31	6	2	2	5 491 522.66	3 661 015.10	1 043 770	3.52
South Carolina	Jan. 1	Mar. 23	2.8	2	Mar. 23	Dec. 31	9.2	3	3	1 511 452.56	546 308.16	127 467	4.28
South Dakota	.....	.....	.....	....	Jan. 1	Dec. 31	12	1	2	624 692.44	624 692.44	131 700	4.74
Tennessee	.....	.....	.....	....	Apr. 1	Dec. 31	9	2	2	812 356.68	541 571.12	173 365	3.12
Texas	.....	.....	.....	....	June 15	Dec. 31	6.5	1	1	1 215 623.36	2 244 227.74	688 233	3.26
Utah	.....	.....	.....	....	Mar. 8	Dec. 31	9.7	2½	2½	404 085.81	200 000.00	59 525	3.36
Vermont	.....	.....	.....	....	Apr. 1	Dec. 31	9	1	1	168 172.81	224 230.41	52 776	4.25
Virginia	.....	.....	.....	....	June 27	Dec. 31	6	3	3	1 556 920.99	1 037 947.32	218 896	4.74
Washington	.....	.....	.....	....	Jan. 1	Dec. 31	12	1	2	1 225 149.66	1 225 149.66	258 264	4.74
West Virginia	.....	.....	.....	....	July 27	Nov. 1	3.1	2	2	366 490.00	709 337.10	157 924	4.50
Wyoming	.....	.....	.....	....	Mar. 1	Dec. 31	10	1	1	140 161.62	168 193.94	39 831	4.22
Total	.....	.....	.....	....	.....	.....	.....	.....	.....	$36 813 939.61	.....	7 440 884	.....
Average	.....	.....	.....	....	.....	.....	.....	.....	.....	.....	.....	.....	$4.46

* Analysis based on Table IV, p. 17, *Good Roads*, April, 1924.

A State highway system contemplating the construction of more than 3 000 miles of roads had been laid out and $18 000 000 (or an average of less than $6 000 per mile) had been provided for the construction of this system.

The Highway Commission and its Chief Engineer undertook to accomplish as much as possible with this small allotment. Having in mind the relatively limited traffic of that day, it adopted as a minimum pavement standard an unsurfaced Portland cement concrete base, 15 ft. wide and 4 in. thick, costing on an average from $6 000 to $8 000 per mile, exclusive of the grading. To-day, pavements on main routes are being constructed not less than 20 ft. wide and 6 in. thick, increasing to 9 in. at the edge, at a cost averaging upward of $25 000 per mile, exclusive of the grading; and yet it can hardly be said with justice that the highway builders of ten years ago should have had sufficient vision to provide in highway construction for the needs of to-day, any more than engineers may be justified at present in looking ten years into the future and building accordingly, especially in view of the fact that insufficient funds have been provided to care for the needs of to-day, let alone those of to-morrow.

The question may well be asked, "Are we justified in looking even five years into the future and in planning accordingly?" The speaker believes so. Referring again to Table 4, the average increase in traffic throughout the United States for the five years, from 1913 to 1918, amounted to 388% and for a similar period, from 1918 to 1923, 146%, while in California during the same periods it was 308% and 170%, respectively. Thus, the rate of increase in California during the last five years was only a little more than one-half that for the preceding five years. It is very doubtful, however, whether the increase during the next five years will amount to even 100 per cent.

With the natural and normal increase which can be expected, and a reasonable increase in the present registration and weight fee, combined with the present 2-cent gasoline tax, the financing of the construction, reconstruction, and maintenance of the California State Highway System should not offer any serious problems.

Dr. Hewes points out the rather high annual maintenance, reconstruction, and improvement costs on State Highway Routes 2 and 4 since the original construction. These costs, however, can hardly be interpreted as indicative of the future. Many of the earlier 4-in. pavements were constructed on poor foundation soils. The combination of the poor soil, the narrow (15-ft.) width, and the exceedingly rapid passenger and truck motor development, put such a severe strain on this light pavement slab as to result in many cases in a rapid deterioration after a few years of service. Some of these sections have been completely resurfaced with from 4 to 5 in. of Portland cement concrete and others with from 1½ to 3 in. of asphaltic concrete, making a total thickness of from 5½ to 8 or 9 in.

At the same time, the width of pavement was increased to 20 ft. in most cases and the old base was thoroughly repaired before resurfacing. The resulting structure is undoubtedly a great improvement on the original con-

struction and is much better able to stand up under modern traffic without undue and rapid deterioration.

Lack of funds has prevented the widening and thickening of much of the system, but many miles of the old 15 ft. by 4-in. bases, where the foundation conditions are good, are still giving excellent service and will undoubtedly continue to do so for some time to come if funds can be provided for the construction of a hard surface shoulder so that traffic can be diverted from its present concentration on the narrower width.

FINAL REPORT OF THE SPECIAL COMMITTEE ON STRESSES IN STRUCTURAL STEEL

Discussion*

By Messrs. D. B. Steinman, Clyde T. Morris, Lewis E. Moore, and J. A. L. Waddell.

D. B. Steinman,† M. Am. Soc. C. E.—The report of the Special Committee on Stresses in Structural Steel‡ is a timely and welcome contribution on a subject of practical interest and importance.

There is always a tendency to perpetuate old standards long after they have outlived their usefulness under changing conditions. In the matter of working stresses for structural steel, engineers have clung too long to an arbitrary standard which came into use when steel was first introduced as a construction material and which represents a needless waste under present conditions. The retention of a working stress that is generally recognized as unnecessarily low, is conducive to looseness in loading assumptions and to other objectionable departures from careful scientific design.

The old concept of the "factor of safety" was an arbitrary blanket factor to cover all the elements of ignorance and uncertainty in the problem of structural design. All the unknown elements of loading distribution, future increase in loading, impact, fatigue effects, inaccuracies of stress analysis, undetermined secondary stresses, strength and uniformity of material, and structural deterioration from neglect, were blindly covered in a single, arbitrary "factor of ignorance". The working stress of 16 000 lb. per sq. in. thus reached, has remained in official specifications and building codes, by inertia, to this day.

If designing practice is to be maintained on a logical and economical basis, in step with the general advancement in the structural art, engineers must abandon the foregoing unbalanced and wasteful method of procedure, and substitute instead a program of evaluating the different elements of uncertainty in the problem of structural design, and then allocate the respective corrections or margins where they belong. Any anticipated load increase should be covered by increasing the assumed live load, not by reducing the basic unit stress. Any impact or shock effect should be covered by adding a proper dynamic increment to the live load stress, not by diminishing the

* This discussion (of the Final Report of the Special Committee on Stresses in Structural Steel, presented at the Annual Meeting, January 21, 1925, and published in March, 1925, *Proceedings*), is printed in *Proceedings* in order that the views expressed may be brought before all members for further discussion.

† Cons. Engr., New York, N. Y.

‡ *Proceedings*, Am. Soc. C. E., March, 1925, Papers and Discussions, p. 392.

basic unit stress. Structural deterioration should be minimized by proper detailing and maintenance, and, where inevitable, should be covered by a suitable increase in the thickness of metal in the affected parts, not by a general reduction in the basic unit stress. Only the uncertainties in the strength and behavior of the material, and such other elements of uncertainty as cannot be allocated elsewhere, should be covered in the margin which determines the basic unit stress.

Even if there had been no improvement in the quality and strength of structural steel, in the accuracy and reliability of stress analysis, in the design of structural members and details, and in the general art of construction, the elimination and re-allocation of many of the elements of uncertainty that were covered by the old "factor of safety", would certainly justify an upward revision of the corresponding old working stress.

With all these considerations taken into account, the report of the Committee is certainly a step in the right direction—the recommended increase of basic unit stress from 16 000 to 20 000 lb. per sq. in. is amply justified. With material the yield point of which is more than 30 000 lb. per sq. in., a working stress of 20 000 lb. certainly leaves a generous margin to cover any elements of uncertainty still remaining. Many engineers have been using working stresses of 20 000 lb. per sq. in., or higher, for their designs whenever arbitrary restrictions did not prevent, and the resulting structures cannot be criticized on the ground of inadequacy.

A generation ago bridge engineers used Cooper's specifications in which the prescribed working stress was 20 000 lb. per sq. in., for dead load and 10 000 lb. for live load. Without dwelling on other considerations, it is sufficient for the present purpose to point out that 20 000 lb. was accepted as a safe working stress for dead load at that time, and it is certainly a safe unit for dead load stress now. The stress that is safe for dead load should be the prescribed basic stress in a modern specification; any other procedure produces unbalanced designs. Any correction for difference between live load and dead load effects should be made in the evaluation of live load stress and dynamic increment.

The Committee's recommendation of 20 000 lb. per sq. in. as a basic unit stress is, therefore, safe and amply conservative. For columns, the Committee has recommended an adequate and consistent reduction from this basic stress, together with a convenient working formula.

The speaker has also read the Minority Report submitted by four members of the Committee, but does not find therein sufficient reason for further postponing action on this long-discussed question. The Committee should be congratulated on presenting this excellent report, and its recommendations should be adopted by the Society.

Clyde T. Morris,* M. Am. Soc. C. E.—In 1918-19, the speaker was associated with Charles Evan Fowler, M. Am. Soc. C. E., in the investigation and report on the Niagara Railway Arch Bridge, which was described by Mr. Fowler in his paper, "Revision of the Niagara Railway Arch Bridge".† At

* Prof. of Structural Eng., Ohio State Univ., Columbus, Ohio.
† *Transactions*, Am. Soc. C. E., Vol. LXXXIII (1919-20), p. 1919.

that time a careful investigation of the permissible unit stresses in structural steel was made. The records of the mill tests of material from which the arch was built about twenty years before were available and after a thorough study of these in connection with strain-gauge measurements of primary and secondary stresses, and their theoretical calculation, it was decided to use unit stresses that accord almost exactly with those recommended in the Majority Report* of the Special Committee on Stresses in Structural Steel. The use of these unit stresses, however, was conditioned on the careful calculation of all stresses, including impact and secondary. Their use in this bridge was further justified by the superior type of details and excellence of both shop and field work in the original construction. F. E. Schmitt, M. Am. Soc. C. E., in his discussion† of Mr. Fowler's paper, remarks, "It does not seem to be extravagant to say that the design was a generation ahead of its time."

The Committee does well in its report to emphasize the importance of having designs prepared by competent designers skilled in proportioning the parts and detailing the connections. Where such conditions exist and where the future possibilities of a change in type or in severity of service has been estimated in selecting the loadings, the use of the unit stresses recommended by the Majority Report are amply justified.

Lewis E. Moore,‡ M. Am. Soc. C. E. (by letter).§—Except in one particular, the writer is in hearty accord with the report of the Committee as it enunciates a conviction that has been growing upon him for some time.

This exception is the stress recommended for flexure, that is, the beam stress. Steel beams are generally designed by using the actual loads, either distributed or concentrated. Concentrated loads are assumed to act at a point, whereas actually they must be distributed over an appreciable part of the length of the beam, as, for instance, a column resting on a beam. The reactions are treated in computation as though they were applied by a knife-edge. The maximum bending moment is then computed, making no allowances for the unavoidable distribution of concentrated loads over a finite area, nor for the negative bending moment produced by the stiffness of end connections, which is actually a not inconsiderable quantity. The design is then made using this maximum bending moment and the working unit stress in bending that exists only on an infinitely thin layer at the extreme top and bottom of the beam. In the case of an I-beam (and somewhat less so in a plate girder), the same quantity of material is used throughout per foot of length, although it is actually working at its full capacity only on a theoretical line extending transversely of the infinitesimally thin outer layer of fibers at the point of maximum moment. For reasons already outlined the actual maximum moment is probably never fully reached even under full design loads. Also, the actual maximum fiber stress can exist only on the outer fibers of the beam, and the adjacent less stressed fibers undoubtedly reinforce them.

* *Proceedings*, Am. Soc. C. E., March, 1925, Papers and Discussions, p. 394.

† *Transactions*, Am. Soc. C. E., Vol. LXXXIII (1919-20), p. 2012.

‡ Cons. Engr., Boston, Mass.

§ Received by the Secretary, January 21, 1925.

The Committee has not hesitated to recommend a working stress of 20 000 lb. per sq. in. for tension members in which each fiber is supposed to carry its full stress; why not then recommend a working fiber stress of 22 000 lb. per sq. in. for beams in which all the fibers are not working at their full capacity and in which the assumptions made in designing are such that the actual computed stress is probably never reached, even under full design loads?

In the report of the Committee, it is stated that "members designed to resist flexure in cross-bending * * * must resist forces of tension, compression, and shear." This is true and is so thoroughly understood, even by the handbook engineer, that it would seem to be an additional argument for the use of higher stresses than would be permissible where the conditions were not so well understood. The writer suggests to the Committee the use of 22 000 lb. per sq. in. in bending where the compression flange is continuously rigidly held.

He is heartily in accord with the Committee in its recommendation that the upper limit of stress in columns be made 16 000 lb. per sq. in. for a slenderness ratio of not more than 50.

The committee of engineers employed by the American Institute of Steel Construction recommended, as is well known, a tensile and bending stress of 18 000 lb. per sq. in. and a column formula based on 18 000 lb., but arbitrarily limited to a maximum of 15 000 corresponding to a slenderness ratio of about 60.

The City of Boston adopted the recommendations of the committee but reduced the maximum stress for columns from 15 000 to 13 500. The writer has designed under this law and finds that with a maximum stress of 13 500 lb. the columns are out of proportion to the remainder of the structure.

J. A. L. Waddell,* M. Am. Soc. C. E.—The speaker was practising as a bridge specialist at the time the transition from wrought iron to steel in bridge construction was being evolved. He drafted some of the first bridge specifications for using carbon steel, and in them specified an elastic limit of 35 000 lb. per sq. in., a working tensile stress of 18 000 lb. per sq. in., and column formulas of $18\ 000 - 70\ \frac{l}{r}$ for fixed ends, and $18\ 000 - 80\ \frac{l}{r}$ for hinged ends.

For a number of years he readily obtained steel having an elastic limit of 35 000 lb. per sq. in., without extra charge; but after a while the steel manufacturers began to cut down the elastic limit until it finally reached as low as 30 000 lb. per sq. in. This was done so as to have a uniform product for all structural purposes and in order to reduce the number of inspectors' rejections to a minimum. Soon after this bad habit was formed by the manufacturers, they refused to furnish the higher grade of steel, on the plea that to do so would involve delay.

It has been shown† that tests for modern bridge steel rarely indicate an elastic limit of less than 35 000 lb. per sq. in.; hence, it would involve no real

* Cons. Engr., New York, N. Y.

† *Proceedings*, Am. Soc. C. E., March, 1924, Society Affairs, p. 263.

hardship for the manufacturers, were they required to furnish bridge metal of that strength.

The increase in strength suggested is about 17%, as compared with standard bridge steel having an elastic limit of 30 000 lb. per sq. in.; hence, theoretically, it would be legitimate to increase the standard intensity of working tensile stress from 16 000 lb. to 18 700 lb. per sq. in.; but, in the speaker's opinion, it is not advisable to stress steel in tension much more than one-half the elastic limit, consequently, he considers that the working-stress intensity for tension should not exceed 18 000 lb.

As for the proper column formulas, there is likely to be a difference of opinion among structural engineers, because of the very prevalent modern opinion that steel struts are not as strong as they were formerly thought to be. The speaker's personal experience in testing struts causes him to disagree; he believes that, in view of the low limits of $\frac{l}{r}$ that very properly are now governing first-class bridge design, it would be perfectly safe to use his old right-line formulas previously mentioned, or some formulas of the Rankine type that would give essentially similar results.

George F. Swain, Past-President, Am. Soc. C. E., has pointed out* that the right-line formula is wholly illogical, while the Rankine formula is the least illogical of all the column formulas that have been used to any extent in practice. The logicalness of a column formula, in the speaker's opinion, cannot be a matter of much moment, in view of the wide divergence of the results of compression tests made on struts built from the same plans, with material from the same melt and rolling, and with as nearly similar conditions of testing as are feasible. It may be true that, for very great values of $\frac{l}{r}$, the Rankine formula is more accurate than the straight-line formula; but in bridge building and other structural steelwork, large values of that ratio are not used, hence, the simpler the formula the more easily is it remembered and applied when one's computing tables and diagrams are not at hand.

In conclusion, the speaker desires to emphasize his previous statement that, in his opinion, the manufacturers of structural carbon steel have no sound reason for refusing to furnish that metal with a minimum elastic limit of 35 000 lb. per sq. in., and without increasing their price therefor.

* In "Structural Engineering."

MEMOIRS OF DECEASED MEMBERS

NOTE.—Memoirs will be reproduced in the volumes of *Transactions*. Any information which will amplify the records as here printed, or correct any errors, should be forwarded to the Secretary prior to the final publication.

SIR MAURICE FITZMAURICE, Hon. M. Am. Soc. C. E.*

DIED NOVEMBER 17, 1924.

Maurice Fitzmaurice was born on May 11, 1861, in County Kerry, Ireland. He was educated at Trinity College, Dublin, obtaining the degrees of B. A. and B. E.

From 1883 to 1885, he was articled to the late Sir Benjamin Baker, Hon. M. Am. Soc. C. E., under whom, in conjunction with Sir John Fowler, he was employed between 1885 and 1888 on construction work on the Forth Bridge. From October, 1888, to June, 1891, he was in responsible charge of the construction of the Chignecto Ship Railway in Canada, and, subsequently, until May, 1892, he was engaged on the designs and estimates for replacing cast-iron bridges on the London, Brighton & South Coast Railway with steel structures.

Later in May, 1892, Mr. Fitzmaurice was appointed Resident Engineer under Sir A. R. Binnie (then Chief Engineer to the London County Council) during the construction of the Blackwall Tunnel.

In 1898, he was appointed Resident Engineer to the Egyptian Government on the construction of the Assuan Dam, the late Sir Benjamin Baker being the Consulting Engineer.

Shortly after his return to England in 1902, and following the retirement of Sir Alexander Binnie, Mr Fitzmaurice was appointed Engineer-in-Chief to the London County Council. The following are some of the works which were designed and carried out by him either entirely or in part, before he retired in 1912: The duplication of the main drainage of London; the London County Council tramways; the Rotherhithe Tunnel; the underground tramway from the Thames Embankment up Kingsway to the north; the Greenwich and Woolwich Footways under the Thames; the Woolwich Ferry jetties; the coal wharf for the tramway power station of the London County Council, at Greenwich; the new Vauxhall Bridge; the prolongation of the Thames Embankment on the north bank up stream from the Houses of Parliament, and the embankment for the new County Hall on the south bank of the Thames. It was in connection with the laying of the foundation stone of the new County Hall that he received the honor of Knighthood.

After resigning his position as Chief Engineer to the London County Council to engage in consulting work, Sir Maurice joined the firm of Messrs. Coode, Son & Matthews, afterward changed, on the retirement of Sir William Matthews, to Messrs. Coode, Fitzmaurice, Wilson & Mitchell, with which firm he remained to the time of his death.

* Memoir prepared by Maurice F. G. Wilson, Esq., London, England.

Among the more important works with which he was associated during this period are: Harbor works at Peterhead, Scotland, Fishguard, Wales, Colombo, Ceylon, Singapore, Straits Settlement, and East and West Africa.

An undertaking to which he devoted much attention and for which his firm acted as Consulting Engineers was the Gezira Irrigation Scheme, which includes the building of the Sennar Dam on the Blue Nile.

In 1913, on behalf of his firm, Sir Maurice visited Australia with a view to giving advise concerning the construction of naval harbors and works; similarly, in 1920, he visited Hong Kong, China, in connection with proposed harbor improvements at that port. In 1912, he was appointed Chairman of the Advisory Committee of the Admiralty on Naval Works, which office he held until 1919.

Associated with Mr. Basil Mott, Sir Maurice was engaged on the following projects: Advising on the question of a further tunnel under the Mersey to connect Liverpool with Birkenhead; designing the new high-level bridge to be constructed at Newcastle-on-Tyne; and advising what was best to be done with Waterloo Bridge when it showed signs of failure.

During the World War, in addition to other activities, he served as Chairman of the War Office Committee on Civilian Labor on the London Defences and as a member of the War Office Committee on Hutted Camps. He twice visited the British Front on the Continent for the purpose of advising on questions of drainage.

During 1918-19 he served as Chairman of the Nile Projects Committee of the Foreign Office and from 1917 to 1919 as Chairman of the Canal Control Committee of the Board of Trade. In 1919, he was appointed Chairman of the Treasury Committee on Aerodrome Accounts. Sir Maurice was a member of the Royal Commission on Fire Prevention, of the Advisory Council of the Science Museum from 1915 to 1921, of the Advisory Council on Scientific and Industrial Research, and of the International Technical Commission, Suez Canal. He was also Colonel Commandant of the Engineer and Railway Staff Corps.

He received the Order of the Mejidieh, 2d Class, in 1901, for his work on the Assuan Dam, and was created a Companion of the Order of St. Michael and St. George the following year. He held the honorary degree of LL.D. of Birmingham University, and had been a Fellow of the Royal Society since 1919. He entered The Institution of Civil Engineers as a Student, became an Associate Member in 1887, was transferred to the grade of Member in 1893, and served as President in 1916. He received for contributions to the Minutes of *Proceedings* of The Institution the Telford and Watt Medals, a Telford Premium, and a Miller Prize. He was also a member of The Institution of Mechanical Engineers, an Honorary Fellow of the Society of Engineers, an Honorary Member of the Royal Engineers' Institution, and a member of the Engineering Institute of Canada.

He was married in 1911 to Ida, daughter of the late Col. Edward Dickinson, R. E., of West Lavington Hill, Midhurst, Sussex, who with two daughters, survives him.

Sir Maurice was elected an Honorary Member of the American Society of Civil Engineers on June 19, 1922.

GEORGE GRAY ANDERSON, M. Am. Soc. C. E.*

Died December 23, 1923.

George Gray Anderson was born in Aberdeen, Scotland, on April 20, 1858. He received his early education at the Robert Gordon's College in his native city. During the years 1872-74, he was a student at the University of Aberdeen, where he won special honors in mathematics and physics. From April, 1874, to April, 1879, he served as an articled pupil with Walker and Beattie, Engineers, of Aberdeen. On the termination of his apprenticeship, Mr. Anderson became an Assistant Engineer on the London-Rugby Division of the of the London and Northwestern Railway in England.

In April, 1880, Mr. Anderson came to the United States and settled in Denver, Colo., where he became Assistant Engineer in the employ of the English company then promoting the Northern Colorado Irrigation Company's project to serve lands adjoining Denver. In June, 1883, he was appointed Chief Engineer and General Superintendent of the Platte Land Company of Denver, which carried out the Northern Colorado Irrigation Company's project and constructed the Larimer and Weld, Loveland and Greeley, and Platte Valley Canals. These projects embraced in all approximately 200 miles of main canals.

In June, 1890, Mr. Anderson severed his connection with the Platte Land Company to become a partner in the prominent consulting engineering firm of Campbell and Green of Denver, the new firm taking the name Campbell, Greene and Anderson. In 1891, on the retirement of Mr. Greene, the firm became Campbell and Anderson and continued under this name until 1896 when the partnership was dissolved.

From 1896 to 1916 Mr. Anderson practiced as a Consulting Engineer with offices in Denver. In 1916, on account of his wife's health, he moved to Los Angeles, Calif., where he continued his consulting practice until his death.

His recognized ability brought him an extensive practice throughout the Western States and in Canada. Among his numerous professional engagements, the following more important ones may be mentioned:

1896-1916, Expert on rates, quality, and pressures, Denver Union Water Company; construction of the Irrigation System of the Alberta Railway and Irrigation Company, Alberta, Canada; investigation and report on Canadian Pacific Irrigation System, Alberta; report on fountain water supply for Pueblo, Colo.; construction of water-works for the City of Greeley, and of Clear Creek Reservoir, Colorado; report on Sacramento Valley Irrigation District, California; appraisal of water rights of Denver Union Water Company; study and report for Dominion of Canada on international issue in St. Mary's River, Alberta *vs.* Montana; Expert in the case of Twin Lakes Placer Company *vs.* City of Pueblo; reports on Laramie Valley Irrigation District, Wyoming, and on Southern Alberta Land and Irrigation Project, Canada; construction of Schaefer Dam and Sanchez Reservoir, Colorado; recon-

* Memoir prepared by the following Committee: Harry Hawgood, *Chairman*, Lyman E. Bishop, F. G. Dessery, and John C. Ulrich, Members, Am. Soc. C. E.

struction of the Otero Canal System and report on the La Plata Land and Canal Company's System, Colorado; examination and report on Buffalo Basin, Riverside and Prewitt Reservoirs, and on a gravity water supply for Grand Junction, Colo.; investigation of weter supply of Spring Valley Water Company, San Francisco, Calif.; study of water rights of hydro-electric properties of Pacific Gas and Electric Company, San Francisco; and appraisal in conjunction with Leonard Metcalf, M. Am. Soc. C. E., of properties of Denver Union Water Company. During this period Mr. Anderson also served as a member of the Board of Consulting Engineers of the Imperial Irrigation District of California.

1917-1923, Investigation and report on power possibilities of the water supply of Colorado Springs, Colo., and for the Government of Alberta on the Lethbridge Northern Irrigation District, and on various projected irrigation districts in Alberta, Canada; and an examination and report on the failure, during the Pueblo flood in June, 1921, of Schaefer Dam, Colorado.

In 1918, Mr. Anderson had a prolonged attack of influenza and in 1923 suffered a second attack, to the serious complications of which he ultimately succumbed.

Interested in irrigation, Mr. Anderson contributed papers to the *Transactions* of the Society on "Irrigation in Colorado"* and, on "The Effect of Alkali on Concrete"†. The Colorado Scientific Society has also published a paper by him on "Some Aspects of Irrigation Development in Colorado"‡. He presented a paper on "Irrigation in Colorado" at the Sixteenth National Irrigation Congress, 1908, and another on "The Combination of Water Resources for Irrigation and Hydro-Electric Purposes," at the Second Pan-American Scientific Congress.

Mr. Anderson was a man of strong individuality, unlimited energy, and tireless industry. During his long residence in Colorado he earned and enjoyed a prestige in his profession unexcelled by that of any confrere. His charming personality gathered to him a host of friends, among whom were some of the most prominent citizens of Colorado, and, in later years, of California, to whom the news of his untimely and unexpected death came as a distressing shock and by whom he will always be remembered with affection and respect.

The younger members of the profession found in Mr. Anderson an approachable, kindly man ever ready to give them counsel in their work or aid them in obtaining employment.

The seriousness with which he accepted his engagements and discharged his responsibilities to the profession is illustrated by his insistence in October, 1923, only two months before his death, in rising from a sick bed in Los Angeles and crossing the Continent, accompanied by his wife, to attend the Fall Meeting of the Society at Richmond, Va. He was able to attend, in a wheel-chair, one meeting of the Board of Direction and thereafter for the remainder of his stay in Richmond was unable to leave his bed. Undoubtedly, this journey, in his weakened condition, hastened his death.

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* *Transactions,* Am. Soc. C. E., Vol. LXII (1909), p. 1.

† *Transactions,* Am. Soc. C. E., Vol. LXVII (1910), p. 572.

‡ *Proceedings,* Colorado Scientific Society, Vol. IX (1908–10), p. 273.

Mr. Anderson was married at Denver, on October 31, 1883, to S. Ella Beck, daughter of Isaac T. Beck, of Denver. He is survived by his wife, two daughters, and a son, Robert Anderson, who is serving in the Flight Research Department of the United States Aviation Service.

He was a member of the Institution of Civil Engineers, the Engineering Institute of Canada, the Colorado Scientific Society, and the Seismological Society of America, and an Honorary Member of Tau Beta Pi. He was also a member of the Denver and Mile High Clubs of Denver, and of the University and Celtic Clubs of Los Angeles.

Mr. Anderson was elected a Member of the American Society of Civil Engineers on February 7, 1906, and, at the time of his death, was a member of the Board of Direction. He took a keen and active interest in the Society's affairs and will long be remembered for his forceful work on the revision of the Constitution into its present form. He was also a Past-President of the Colorado and Los Angeles Sections of the Society.

JOHN CUMMINGS AUTEN, M. Am. Soc. C. E.*

DIED OCTOBER 15, 1923.

John Cummings Auten, the son of Wesley and Mary Nelson Auten, was born at Montandon, Pa., on August 31, 1870. He was of Huguenot descent and spent much time and labor collecting the history of his family.

In August, 1887, Mr. Auten entered the service of the Pennsylvania Railroad and remained with that Company all his life. He filled, successively and successfully, the following positions: August 5, 1887, Rodman in the Engineering Corps of the Sunbury Division; May 1, 1894, Assistant Supervisor on the Sunbury Division; November 1, 1894, promoted to the position of Supervisor of that Division; October 1, 1898, transferred to the Baltimore Division and on December 10, 1901, to the Middle Division. On April 10, 1902, he was promoted to the position of Assistant Engineer of the Elmira and Canandaigua Division and, on June 1, 1903, was transferred to the West Penn Division where he served until April 1, 1907, when he was transferred to the Eastern Pennsylvania Division. On November 16, 1908, he was transferred to the West Jersey and Seashore Railroad and on January 15, 1910, was appointed Division Engineer of the Maryland Division. On February 11, 1914, Mr. Auten was promoted to the position of Principal Assistant Engineer of the Philadelphia, Baltimore and Washington Railroad, in which position he served until December 16, 1919, when he was transferred to the office of the General Superintendent of the Southern Division at Wilmington, Del., and on May 1, 1920, to the Baltimore Division, to which Division he was attached at the time of his death.

Mr. Auten was always fair and considerate with those under him and his genial disposition made him a favorite with his associates. He was honored

* Memoir prepared by Franklin Duane, Esq., Wilmington, Del.

and esteemed by his superior officers. In private life he was a devoted husband and father.

On April 23, 1916, Mr. Auten was installed and ordained an Elder of Westminster Presbyterian Church, Wilmington, which honorable position he held until his death. He was an enthusiastic and devoted member of the Order of Masons, being a member of the Blue Lodge, the Scottish Rite, and St. John's Commandery of Wilmington, and a member of the Shrine, Lulu Temple, Philadelphia, Pa. He was also a member of the City Club and the Whist Club of Wilmington.

On October 14, 1904, Mr. Auten was married to Ella Hawkins, of Parkton, Md., who, with two children, John Hawkins and Elizabeth, survives him.

Mr. Auten was elected a Member of the American Society of Civil Engineers on May 13, 1918, and took great interest in its activities.

WALTER FRANCIS BALLINGER, M. Am. Soc. C. E.*

Died December 23, 1924.

Walter Francis Ballinger was born in Petroleum Center, Venango County, Pa., on August 13, 1868, and received his education at the Woodstown, N. J., Grammar School; the Philadelphia, Pa., Business College; the International Correspondence School, Scranton, Pa.; and Drexel Institute, Philadelphia, Pa.

Mr. Ballinger entered the architectural offices of Geissinger and Hales, in Philadelphia, in 1889, and in 1895 became a member of the firm which was continued under the name of Hales and Ballinger. He became the senior member of the firm under the name of Ballinger and Perrot in 1901, which firm in 1920 became The Ballinger Company.

He became interested in the possibility of reinforced concrete building construction when there were only a few such buildings in existence and he did much to aid in the development of this form of construction. He was also co-inventor of a new type of saw-tooth roof construction known as "Super-Span", which while it possesses the advantages of the natural illumination of that type, eliminates a large percentage of the columns usually needed.

Mr. Ballinger took an active interest in professional societies in connection with architecture and engineering. He was a member of the National Fire Prevention Commission, the American Society of Mechanical Engineers, Franklin Institute, Engineers Club of Philadelphia, Engineers Club of New York, Pen and Pencil Club, and the Society of Industrial Engineers.

Having been interested in all organizations for civic improvement, he was a member of the Philadelphia Chamber of Commerce, the Philadelphia Board of Trade, the Germantown and Chestnut Hill Improvement Association, the Fairmount Park Art Association, and the City Club of Philadelphia.

Mr. Ballinger was registered as an Architect and Professional Engineer in the States of Pennsylvania, New York, and New Jersey. His professional practice included the design and supervision of erection of commercial and

* Memoir prepared by William R. Fogg, Secy., The Ballinger Co., Philadelphia, Pa.

industrial plants, institutional buildings, including hotel, hospital, and club buildings, as well as churches and schools.

As a member of the Methodist Episcopal Church, Mr. Ballinger was active in its work, as well as in that of local charitable organizations, having been a member of the Board of Temperance, Prohibition and Public Morals and of the Philadelphia Missionary Society of this Church. He was also a member of the Philadelphia Law Enforcement League and of the Board of Managers of the Pennsylvania Seamen's Friend Society and the Seamen's Church Institute.

He was a Mason, a member of the Blue Lodge, Chapter, Consistory, and Shrine, and, in addition, was a member of the Manufacturers Club, Penn Athletic Club of Philadelphia, North Hills Country Club, Lake Placid Club, and the Young Men's Christian Association.

Mr. Ballinger died in Philadelphia, on December 21, 1924, as the result of an automobile accident.

In 1896, he was married to Bessie M. Connell, who, with one daughter, Grace A., now engaged in social service work, and one son, Robert I., a member of The Ballinger Company, survives him.

Mr. Ballinger was elected a Member of the American Society of Civil Engineers on February 7, 1906.

HENRY HARRISON FARNUM, M. Am. Soc. C. E.*

October 23, 1924.

When Henry Harrison Farnum at the age of eighty passed from earth on October 23, 1924, he left behind him but three members of his class of 1865 and perhaps not more than sixteen or seventeen of Rensselaer's Alumni who had been graduated before that time. There are, therefore, few to tell the history of those early years.

Mr. Farnum's paternal ancestor, Henry Farnham, came from Old England to New England in 1638. His mother's ancestor, Jacob Caudebec, of French Huguenot stock, left his home, in Caudebec, France, shortly after the Revocation of the Edict of Nantes, reaching a permanent home in New York State about 1690. The subject of this brief sketch may well be called an American.

Many representatives of the family had chosen the profession which Mr. Farnum made his own, and it was, therefore, quite in keeping that in 1865, at the age of twenty-one, he should have completed the course in Civil Engineering at Rensselaer Polytechnic Institute, Troy, N. Y., entitling him to the degree of Civil Engineer. While at Rensselaer he became one of the founders of the Theta Xi Fraternity, a strictly engineering organization with twenty-seven Chapters located at engineering schools and universities having engineering courses. In this fraternity he took a hearty interest and was always proud of his connection as one of its founders.

* Memoir compiled under the direction of William H. Wiley, M. Am. Soc. C. E.

His boyhood and early youth were spent in Port Jervis, N. Y., where one of the lineal descendants of Jacob Cuddeback—Asenath Cuddeback—had become the wife of Samuel B. Farnum, a descendant in direct line from Henry Farnham, both family names being Americanized. Their fourth child was Henry Harrison. At the age of fourteen, he was sent to Flushing Institute, then presided over by Elias Fairchild, of whom his former pupil could never speak too highly.

So few remain who knew Mr. Farnum during his life in Troy that it seems not inappropriate to quote a few sentences penned by one who was his friend there, and whose friendship has continued to this day. "A most lovable friend," the letter says, "kind and sympathetic. A fine scholar. His sympathy included all who needed his help and it was always cheerfully given."

The years from 1866 to 1868 were spent in mining engineering in Colorado. He crossed the plains before the Union Pacific had made its way over, and he sometimes spoke of that journey of eight days and nights in a Concord coach with Indians lurking near enough at hand to make both driver and passengers continually alert.

After his return from Colorado, Mr. Farnum became Assistant Engineer on the preliminary surveys and location of the Port Jervis and Monticello Railroad in New York. During 1869-70, he served as Assistant Engineer for the New Orleans, Mobile and Chattanooga Railroad, engaged in surveys for location and construction, especially for the westerly extension.

From 1870 to 1875, Mr. Farnum was Chief Engineer for the South Side Railroad of Long Island. He lived then in New York, or in Brooklyn, N. Y., and his days were given to engineering, his evenings to the studious enjoyment of the best that New York could offer to satisfy his ardent love for the drama and music, particularly the latter. Theodore Thomas was then a great figure in the musical life of the city, and the young engineer derived infinite pleasure and profit from the gratification of his fine taste for the best.

After finishing his work with the South Side Railroad, private and miscellaneous engineering engaged Mr. Farnum's attention until 1881, and he was for several years engaged with experimental investigations in the process of refining iron, embracing the designing and construction of an iron plant at Chicago, Ill., including heavy machinery and a very heavy foundation.

In 1890, he returned to New York and entered the service of the City in what is now the Borough of the Bronx, and, from that time, he was engaged almost continuously in this Borough, first as Assistant Engineer, then as Chief Engineer of Sewers. At the time of his retirement from active service, about 1923, after many years of profound and intelligent interest in the phenomenal growth and development of the Borough, he was still occupied in the exercise of his professional skill toward making that development healthful.

The period from 1890 to 1901 covers the development of the sewer system in this Borough. It was here Mr. Farnum's fortune to build the largest and longest sewer in the United States. Owing to the varying natural conditions of the drainage distribution of the extensive marsh lands bordering on the Harlem River, the East River, and Long Island Sound, new problems were

presented in the case of each outlet sewer, in order to secure a permanent foundation. This task was successfully accomplished.

The closing years of Mr. Farnum's earthly life were greatly enriched in affording more leisure for historical reading which was always his delight, in giving time for home pursuits which were always dear to him, and in following with great interest the developing lives of the younger members of the family. There were many on both sides of the house who bear joyful witness to the stimulus of his influence.

He was married, in 1887, to Elizabeth Beattie, who, with his son, Dr. Waldo B. Farnum, his daughter-in-law and two grandchildren, survives him.

Mr. Farnum was a member of the Presbyterian Church from early manhood. As son, brother, husband, and father, in all the most intimate relationships as well as in those more widely known, his life has spoken, and those who knew him in each phase of his long and varied career will bear glad tribute to his worth. One who knew and cherished him wrote, "A gentleman, gallant and unafraid, has passed on." Permit one added word, "A Christian gentleman, gallant and unafraid, has passed on."

Mr. Farnum was elected a Member of the American Society of Civil Engineers on July 1, 1891.

HOWARD SOULE, M. Am. Soc. C. E.*

Died December 10, 1924.

Howard Soule was born on December 8, 1829, at Sennett, Cayuga County, N. Y. He received a common school and academic education at Auburn, N. Y.

In his early experience Mr. Soule was engaged in making preliminary surveys for railways from Auburn to Ithaca, N. Y., and from Auburn to Lake Ontario; later, in 1856, 1857, and 1858, he was in charge of the construction of the railroad from Auburn to Lake Ontario, which is now part of the Lehigh Valley System.

Between 1847 and 1874 he was engaged principally on State projects in connection with the enlargement of the Erie Canal in Central New York. He made surveys for draining the Montezuma Marshes at the foot of Cayuga Lake and was in charge of the work done from 1858 to 1861 in consummation of that project. In 1862 and 1863, he made the surveys for the proposed "Gun-Boat" locks on the Middle Division. He served the State in the positions of Assistant Engineer, Resident Engineer, and Division Engineer, of the Middle Division of the State Canals, ending his State service in 1874.

Mr. Soule served as City Engineer of Syracuse, N. Y., from 1874 to 1876. Afterward, for twelve years, he was engaged in the contracting business, designing and constructing iron bridges, of the Whipple, cast-iron, bow-string type, in general use for many years on the State Canals. At other times during this period he was engaged in the construction of seventy miles of the

* Memoir prepared by Henry C. Allen, M. Am. Soc. C. E.

"West Shore" Railroad, in raising the banks and locks of the Welland Canal (in Canada), and in lengthening the locks of the Erie Canal.

In 1890, he was called on to act as Consulting Engineer and to direct the design for the installation of a supply of water to Syracuse from Skaneateles Lake. He continued in that capacity until the water commenced to flow through the first conduit on July 4, 1894.

Although in practical retirement after this time, Mr. Soule's counsel was often sought, and, in 1905, notwithstanding his advanced age, he again gave valuable service in the installation of the second conduit from Skaneateles Lake.

Howard Soule was a skillful engineer and retained his practical ability and keen common sense to the end of his days. Self-effacing, and retiring in disposition, he holds a high place in the esteem of those who were fortunate enough to know him.

Mr. Soule was elected a Member of the American Society of Civil Engineers on March 17, 1869.

BENJAMIN EMANUEL WINSLOW, M. Am. Soc. C. E.*

DIED NOVEMBER 24, 1924.

Benjamin Emanuel Winslow, the son of Christian Wilhelm and Christiane Winslow, was born in Chicago, Ill., on July 2, 1867. After the Chicago fire of 1871, his parents removed to Richmond County, Virginia, where he spent his boyhood and was educated. In 1878 his family went to Denmark, and his education was continued at Copenhagen where from 1878 to 1883, he attended Hauck's Latin og Real Skole, from 1883 to 1888, the School of the Technical Society, and from 1889 to 1891, the Royal Academy of Fine Arts.

Mr. Winslow then returned to the United States, and on April 9, 1897, was appointed Architectural Engineer under the Commissioner of Buildings, City of Chicago, where he continued until 1906. From that date to 1915, he was employed as Architectural Engineer under several architects and engineers, notably Holabird and Roche of Chicago. In 1915, he was re-appointed to his former position as Architectural Engineer under the Commissioner of Buildings, City of Chicago, passing on structural plans submitted by applicants for building permits. He remained in this position until his death, except for a brief period, 1917-19, when he was assigned as Architectural Engineer in the Architect's Office of the Board of Education, Chicago.

Mr. Winslow was the inventor of the "Winslow Slide-Rule" and the author of the "Winslow Tables" for calculating the strength of wood, steel, and iron beams and columns. He was an engineer of ability and a man of exceptionally high standards of integrity and professional conduct, respected and esteemed by all who knew him.

He was married on October 18, 1908, to Hilfred Rudbeck, who, with one daughter, Gerda Marion, survives him.

* Memoir prepared by T. L. Condron, M. Am. Soc. C. E.

Mr. Winslow was elected a Member of the American Society of Civil Engineers on October 2, 1907. He was also a member of the American Institute of Architects.

ERNEST DRINKWATER, Assoc. M. Am. Soc. C. E.*

Died December 30, 1924.

Ernest Drinkwater was born in Manchester, England, on January 22, 1869. Both his father, Thomas Holbrook Drinkwater, and his mother, Lucy (Dutton) Drinkwater, came of old Cheshire stock, tracing their descent to a period before the Norman Conquest.

He received his education in the elementary and technical schools of his native city, and in 1887 apprenticed himself to his father to learn the business of general contracting. At the termination of his apprenticeship, he became Assistant Engineer for his father's firm, and in August, 1896, assumed control of it, conducting the business successfully until 1902. Mr. Drinkwater had a natural talent for municipal work, and most of the firm's operations were along the lines of estate development, paving, sewerage and water supply, and housing.

In 1902, or early in 1903, he became convinced that better opportunities for professional advancement existed in the Dominions than in the "Old Country." Accordingly he emigrated to Canada (as did many other young Englishmen of his generation), settling in Montreal, Que., in the vicinity of which he spent the remainder of his life.

From May, 1903, to June, 1909, Mr. Drinkwater was in the employ of the Montreal Tramways, first as Construction Foreman, then, successively, as Assistant to the General Construction Superintendent, Inspector on powerhouse and sub-station construction, Storage Battery Engineer, and in charge of electrolysis surveys.

In July, 1909, he was appointed Superintendent of Construction and Chief Inspector of Works for the City of Westmount, a Montreal suburban municipality, a position which he held until June, 1912. The population of Montreal was increasing by leaps and bounds at that time, which increase required the expenditure of much more than the average amount of funds in street paving, sewers, etc.

On the termination of his Westmount engagement, Mr. Drinkwater became Engineer of the Town (now City) of St. Lambert, Que., just then beginning to shake off the lethargy of years of non-activity. He found it a quiet country village, with nothing in the way of modern municipal improvements save a water supply and sewerage system of sorts. He left it an up-to-date suburban residential city, doubled in population and lacking in nothing required for the health, comfort, or convenience of its citizens. During his comparatively

* Memoir prepared by R. De L. French, M. Am. Soc. C. E.

short tenure of office, practically all settled streets were permanently paved and provided with sidewalks, the water supply and sewers were extended to care for the growing population, alterations and additions were made to various municipal buildings, the street lighting system was completely overhauled and modernized and a number of parks were acquired, graded, and planted.

In December, 1916, Mr. Drinkwater opened an office in Montreal as a Consulting Municipal Engineer, which he maintained until 1920, when he was again called to St. Lambert as City Engineer, serving until 1923. While practicing independently, he designed and constructed the water supply and sewerage systems for the Town of Montreal South. In 1923, he again took up consulting practice, in which he continued until his death.

Among Mr. Drinkwater's tasks, during his second term of office in St. Lambert, was directing the expenditure of about $750 000 under the Provincial Housing Scheme. Upward of one hundred and fifty moderately priced residences were built, which materially helped to relieve the shortage of dwellings due to war conditions. It has been stated that this was one of the few projects under the Provincial Law to be carried out with entire success, as that Law did not work as well as its promoters hoped.

One of Mr. Drinkwater's outstanding characteristics was his ability to find time, despite his many professional duties, to take an active part in any social or charitable movement. Thus, he served as Honorary Secretary of the St. Lambert Soldiers' Memorial Committee, Honorary Treasurer of the St. Lambert Club, Vice-President of the Canadian Christmas Tree League, a Life Member of the Canadian Red Cross Society, and a Member of the Canadian Club of Montreal, the St. Lambert Players' Club, the St. Lambert Operatic Society, the Navy League of Canada, the Aerial League of Canada, the Royal Arcanum, the Sons of England Benefit Society, and St. Lambert Lodge, A. F. and A. M.

Mr. Drinkwater was also a member of the American Water Works Association, the American Society of Municipal Improvements, the American Public Health Association, the Royal Institute of Public Health, The Town Planning Institute of Canada, the South Shore Joint Town Planning Board, the Canadian Good Roads Association, and a Fellow of the American Geographical Society.

Having been something of an athlete in his younger days—he had won a number of medals at cross-country running—he never lost his interest in sport, although too busy a man to devote much time to it himself; he claimed that keeping up with his profession gave him sufficient exercise. He was a member of the Country Club of Montreal, the Royal Automobile Club of Canada, the St. Lambert Boating Club, and the Canadian Canoe Association.

In 1904, Mr. Drinkwater was married to Jennie Moores, of Manchester, England, who, with three children, Eric, Mollie, and Hereward, survives him.

Mr. Drinkwater was a man of strong opinions and principles, and had no hesitation in fighting for what he thought to be right. Although he had

multifarious interests, he managed to keep in touch with the affairs of all the organizations of which he was a member, and could be thoroughly depended on in a crisis. He will be sadly missed by his professional brethren, and his death leaves a gap which those associated with him in social and charitable work will find hard to fill.

Mr. Drinkwater was elected an Associate Member of the American Society of Civil Engineers on September 11, 1917.

JOHN WARREN DuBOIS GOULD, Assoc. M. Am. Soc. C. E.*

Died September 20, 1924.

John Warren Dubois Gould was born at White Plains, N. Y., on July 29, 1881. He was the son of the late E. Sherman Gould, M. Am. Soc. C. E., a well known engineer, and a great-grandson of Judge James Gould who founded, in a log house at Litchfield, Conn., the first law school in America. Mr. Gould's earlier education was obtained in the public schools of Yonkers, N. Y., in the St. John's Military School, of Manlius, N. Y., and in the Dwight School of New York City. In 1898, he entered the College of Engineering of New York University, from which he was graduated in 1902, with the degree of B. S. Three years later, in 1905, following experience in the field, he earned the degree of C. E. He was a member of the Psi Upsilon fraternity, and active in behalf of his Alma Mater both as an undergraduate and as an alumnus.

Immediately after his graduation, Mr. Gould went to the Far West and engaged in railroad surveys and maintenance, and, later, in construction work for the Bay Cities Water Company, of San Francisco, Calif., especially in connection with the problem of additional water supply. In 1904, he became Assistant Engineer for the United States Reclamation Service in building the Gunnison Tunnel at Montrose, Colo. From 1904 to 1906, he was associated in New York, N. Y., with the Committee on Fire Prevention of the National Board of Fire Underwriters, as Engineer in charge of reports on the municipal water supply and distribution systems of important cities, including New York and Brooklyn, N. Y. From August to December, 1906, he was engaged as Principal Assistant Engineer of the Passaic River Flood District Commission, preparing surface estimates and reports on a proposed system of flood control for Paterson and Passaic, N. J., to be completed at an estimated cost of $3 000 000. This report has just been presented to the State Legislature of New Jersey.

From 1907 to 1911 Mr. Gould was connected with the firm of Gunn, Richards and Company. Previous to this, his work had been mainly in the technical problems of engineering, but from this time on, his experience became

* Memoir compiled by Dean A. L. Bouton, Coll. of Arts and Science, New York Univ., New York, N. Y.

such as to require, in addition to technical qualifications, executive ability, the power of handling men, and of dealing with individuals representing important interests. During this period he undertook investigations and made reports on various industrial properties and on engineering and other projects, both for private corporations and for the U. S. Department of the Interior, under Secretary Garfield. He was then appointed Special Indian Inspector, and did much useful work for the Government, on the basis of his inspection of Indian Reservations in Colorado, and other Western States.

In 1911, Mr. Gould entered private engineering practice with an office in New York, and was engaged from that time until his death in problems of organization and re-organization. In 1912, he went abroad to carry on negotiations in behalf of the Indian Refining Company. From 1912 to 1914, he was engaged in closing the automobile manufacturing activities of the American Locomotive Company. In 1915-16, he visited Haiti and prepared an important general report on conditions in the island, in behalf of Messrs. Breed, Elliott and Harrison, of Cincinnati, Ohio, and Messrs. P. W. Chapman and Company, of Chicago, Ill., and New York. His report, made as a Special Representative in connection with the Chartered Company of Haiti, was of great value to the American Government and received the especial approbation of President Roosevelt.

During the World War, from July, 1917, until 1919, he was employed with the Food Administration as an Executive Assistant under Herbert Hoover, Hon. M. Am. Soc. C. E., and after the Armistice was in charge of the demobilization of the great organization which war conditions had brought into being.

For almost three years, beginning in 1919, Mr. Gould was at the head of the Industrial Department of Messrs. Bonbright and Company, but, in 1922, he returned to private engineering practice, in which he continued until November, 1923, when he accepted the large responsibilities of the General Managership of the H. H. Franklin Manufacturing Company, of Syracuse, N. Y., a position in which he continued until his resignation on February 1, 1924, on account of illness.

Mr. Gould's career was cut off when it was still incomplete and apparently at the moment when its supreme fruition in achievement was just beginning, but not before the qualities of a rare character were fully defined. Thoroughness in every detail; profoundly sympathetic but accurate judgment of men; an imaginative grasp of great projects; unusual powers of organization and of execution in large business problems, together with the ability to secure the loyal co-operation of subordinates; vital intensity of devotion to any responsibility which he had once accepted, joined with a fine capacity for intellectual detachment, by which he was able to preserve his sense of proportion; modesty and great generosity of spirit; and fearless and inflexible integrity in behalf of the highest business and personal standards, all these qualities make an unusual personal heritage to his profession, and a rich memory of unforgettable inspiration to his family, and to all who had the privilege of his personal friendship.

He died suddenly at his summer home, in Sharon, Conn., of a thrombus in the heart, on September 20, 1924, and is survived by his wife, Evelyn L. Fisk, daughter of the late Harvey Fisk, and two children, John and Evelyn.

Mr. Gould was elected a Junior of the American Society of Civil Engineers on October 7, 1902, and an Associate Member on October 1, 1913.

EDWARD GILBERT MATHEWS, Assoc. M. Am. Soc. C. E.*

DIED JULY 19, 1924.

Edward Gilbert Mathews, the son of Tertullus G. and Sarah A. Mathews, was born in Brooklyn, N. Y., on December 30, 1883. He attended Adelphi Academy, in Brooklyn, later going to Riverview Military Academy, at Poughkeepsie, N. Y., where he spent three years in preparation for Princeton University. He entered Princeton in the fall of 1903, and was graduated in 1907 with the degree of Civil Engineer.

Mr. Mathews' first position was with the Hennebique Construction Company, on the design of reinforced concrete structures. In 1909, he entered the employ of the United Fireproofing Company, Engineers and General Contractors, specializing in reinforced concrete construction. Later, he became Vice-President of this Company, which position he held until 1913, when he was elected President. In 1918, Mr. Mathews was retained by the Taintor Company as Consulting Engineer in connection with certain additions to the manufacturing plant. In 1919, he was retained permanently by the Taintor Company, at the same time being Vice-President and Director of the United Fireproofing Company. On February 4, 1920, he was elected a Director of the Taintor Company.

In January, 1924, he resigned as Vice-President of the United Fireproofing Company, to accept the position of Treasurer of the firm, still remaining actively engaged with the Taintor Company.

He was held in high esteem by his many friends and business acquaintances. He was a very able engineer and organizer, showing promise of reaching an even higher position in his profession. He also took a great interest in the activities of the communities in which he resided.

He was a member of the Princeton Engineering Association, the Princeton Club of New York, and the Princeton Elm Club.

Mr. Mathews and Miss Estelle Smith were married on April 20, 1908, in Brooklyn, where they made their home. Mr. Mathews' death occurred at his summer home in Hartsdale, N. Y. Besides his widow and mother, he is survived by his two children, Frances and Gilbert.

Mr. Mathews was elected an Associate Member of the American Society of Civil Engineers on June 24, 1914.

* Memoir prepared by Messrs. James Dusenbery and Lemuel Osborne.

LEROY TALLMAN, Assoc. M. Am. Soc. C. E.*

Died November 30, 1924.

Leroy Tallman was born at Portsmouth, R. I., on October 28, 1873. He obtained his elementary education in his native town, and was graduated from the Fall River High School, Fall River, Mass.

From June, 1893, to September, 1897, Mr. Tallman served as Rodman, Instrumentman, and Chief of Party with Wolstenholme and Buffinton, Civil Engineers, of Fall River. From October, 1898, to April, 1900, he was with E. I. Marvell, Civil and Mill Engineer, as Engineer in charge of the office at Fall River.

From April, 1900, to September, 1904, Mr. Tallman was engaged as Assistant Engineer in charge of the design and construction of reservoirs and a filter system for the Newport Water Company, Newport, R. I.

In April, 1905, he joined the forces of the O'Rourke Engineering Construction Company as Superintendent of Construction on the Pennsylvania Railroad Tunnels under the Hudson River, from 33d Street, New York, N. Y. On the completion of the tunnels, he was transferred to work on the caisson foundations of the Hudson Terminal Buildings, acting as Assistant Engineer. He was then given charge of the building of the important project known as the Baltimore Sewerage System, as Superintendent for M. A. Talbot Company, and working under the direction of Mr. Calvin Hendrick.

Mr. Tallman took charge thereafter of the construction of the Degraw Street Tunnel, connecting Gowanus Canal with Gowanus Bay, New York, representing the John Pierce Company. On the completion of this work, he joined the organization of Snare and Triest, and built the Long Island City approaches to the Queensboro Bridge. Later, he directed harbor and sea-wall work at Michigan City, Ill.

On his return to the East in the spring of 1910, Mr. Tallman became connected with the Pittsburgh Contracting Company, as Superintendent on Contract No. 52 on the Elmsford Aqueduct, Elmsford, N. Y. This Company was a subsidiary of Booth and Flinn, Limited, with which firm Mr. Tallman continued until his death. He had been, therefore, a member of the Booth and Flinn organization for fourteen years, during which time he was in charge of the following construction projects: Several sections of the Passaic Valley Sewerage System, most of which was through soft ground and under compressed air; the Clark Street Tunnel for the Transit Commission, City of New York; several sections of sewers for the Borough of Brooklyn, City of New York; a number of important tunnel sewer contracts for the cities of Detroit, Mich., and Toledo, Ohio; and, at the time of his death, Contract No. 4 of the Hudson River Vehicular Tunnels (the under-river portion at the New Jersey side).

Mr. Tallman is survived by his wife, having no other near relatives.

* Memoir prepared by S. M. Rutledge, Esq.

He was a member of Eureka Lodge, F. and A. M., and of Portsmouth Royal Arch Chapter, and was President of the Malba Field and Marine Club, Malba, Long Island.

Mr. Tallman was elected an Associate Member of the American Society of Civil Engineers on February 2, 1909.

JOHN BAPTISTE LOBER, Affiliate, Am. Soc. C. E.*

DIED DECEMBER 21, 1924.

John Baptiste Lober was born in Camden, N. J., on April 11, 1848. He was educated in the public schools of Camden, and Pierce Business College, Philadelphia, Pa.

In 1870, he entered business with the firm of Warren Kirk and Company of Philadelphia, manufacturers of coal-tar products. Mr. Lober and Mr. E. Burgess Warren re-organized this Company and continued the business as the Warren-Lober Company until 1883. In that year the Warren-Lober Company was purchased by Mr. M. Ehret, backed by Messrs. W. L. Elkins and P. A. B. Widener, and the name was changed to the Warren Ehret Company, with Mr. Lober as President. Although organized as a roofing business, it soon became engaged in concrete work, particularly, sidewalks and floors. In view of the Company's success in this work, a consolidation proposal was offered by the Vulcanite Paving Company and accepted in 1890. Mr. Lober became Vice-President of the consolidation which retained the name of the Vulcanite Paving Company. This firm, one of the largest of its kind in the United States, used large quantities of imported German cement; at that time there was no Portland cement in the United States sufficiently uniform in character for use by the Vulcanite Paving Company which had established a reputation for superior quality of work.

Business increased to such an extent that it became advisable for the Company to manufacture its own cement. It acquired a tract of land in Warren County, New Jersey, containing a high grade of cement rock and, in 1894, its owners organized the Vulcanite Portland Cement Company, of which Mr. George W. Elkins became President, Dr. A. B. Bonneville, Vice-President, and Mr. Lober, Secretary and Treasurer. A cement mill was erected with two 40-ft. kilns and a daily capacity of 125 bbls. From the organization of the Company in 1894, Mr. Lober was its active Manager. He recognized the need for a domestic cement which would be equal in quality to the best foreign brand. The first 1 000 bbls. were rejected by Mr. Lober as not satisfactory, but, thereafter, the product met all the requirements of the paving companies and others engaged in similar business.

In 1903, Mr. Lober resigned as an officer of the Vulcanite Paving Company to become President of the Vulcanite Portland Cement Company, with which he served actively until a few months before his death at Wynnewood, Pa., on December 21, 1924.

* Memoir prepared by Albert Moyer, Affiliate, Am. Soc. C. E.

Mr. Lober was a leader in the cement industry. He was one of the organizers of the Portland Cement Association in 1902. He served as a second President of the Association and as its President for nine years. He was Treasurer of the Eastern State Institution for the Feeble Minded and Epileptics at Spring City, Pa., and a member of the Union League Club of Philadelphia, and the American Society for Testing Materials.

He had charge of large construction and manufacturing projects and was a highly talented, honorable, upright, and just business man. The memory of his achievements will always be an inspiration for others.

One of the last contributions which Mr. Lober made to the Portland cement industry was the book, "History of the Portland Cement Industry in the United States", written in collaboration with R. W. Lesley, Affiliate, Am. Soc. C. E., and Mr. George S. Bartlett. The work covered a period of nearly two years and included personal reminiscences, data, and historical facts relating to the development of the Portland cement industry in the United States.

Mr. Lober is survived by his widow, Mrs. Clara Lober, and a son, William D. Lober.

Mr. Lober was elected an Affiliate of the American Society of Civil Engineers on July 9, 1906.